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DESCRIPTION AND TRIALS OF TORPEDO-BOAT DESTROYERS *FLUSSER* AND *REID*.

BY CAPTAIN WILLIAM NELSON LITTLE, U. S. N.

The Congress, by its Acts making appropriations for the naval service for the fiscal years ending June 7th, 1907, and June 30th, 1908, provided for the construction of five torpedo-boat destroyers, to be built in accordance with certain provisions contained in an Act of August 3d, 1886, and to be in all their parts of domestic manufacture. Accordingly, after the customary advertisement, contracts were made with the Bath Iron Works, Ltd., of Bath, Maine, for the construction of two torpedo-boat destroyers designated by the numbers *20* and *21*, afterwards named the *Charles W. Flusser* and the *Reid*.

The contract was signed September 28th, 1907, and called for completion and delivery within two years from date of contract. Delivery of the *Flusser* was made to the Commandant of the Navy Yard, Boston, Mass., September 29th, and of the *Reid* October 27th, 1909.

Both vessels were built as specified, with very few and only minor changes, and, excepting in case of the *Reid*, which was one month over due, they were built without vexatious

delays. When one considers the latitude of Bath and the corresponding temperatures of its winters, and the further fact that these vessels were built out in the open, some credit to the sturdiness of the artisans, helpers and others who wrought so well is impelled.

As to the contract requirements of these vessels, the following are some of the particulars: Displacement, 700 tons; speed, 28 knots; to be built on plans furnished by the Navy Department as to type, armament and equipment; the machinery to be Parsons turbines capable of propelling the vessel at the specified speed with an expenditure of steam, as follows:

At full speed, 15.5 pounds of water per shaft horsepower an hour.

At 24 knots for 12 hours, 16.5 pounds of water per shaft horsepower an hour.

At 16 knots for 24 hours, 25.2 pounds of water per shaft horsepower an hour.

The following trials were required:

A standardization trial over a measured-mile course in deep water, to ascertain the revolutions of the screw propellers corresponding to the measured speeds, from which to construct the speed and revolution curve of the ship, shown at the end of this article.

A full-speed trial of four hours' duration, with only the necessary auxiliary machinery in operation and a fireroom air pressure limited to five inches by water, and the steam pressure at the H.P. turbine not to exceed 240 pounds by gauge.

A twelve-hour trial at a nearly constant speed of twenty-four knots, the coal and steam consumption to be measured.

A twenty-four-hour trial at sixteen knots, during which the coal and steam consumption was to be measured.

The contract price for each of these vessels was six hundred and twenty-four thousand dollars, of which amount two hundred and fifty-three thousand was for the hull, foundations for and installation of armament, and boat and ordnance out-

fit complete by the contractors after the last two items shall have been supplied by the Government, and three hundred and seventy-one thousand dollars for the steam machinery under cognizance of the Bureau of Steam Engineering. Which is to say that the machinery cost is to the hull, equipment and ordnance (installation work) as six to four (nearly). The contract price included values for spare parts supplied, as usual, by the contractors.

GENERAL DESCRIPTION OF THE VESSEL.

HULL.

The hull is built of medium and hard steel, galvanized in all parts of shell and frame below the water line, and all parts lightened by omitting or by cutting out useless material. The proportions of length to beam of the hull are as 11.5 to 1, nearly.

There is a main deck and raised forecastle, the latter extending as far aft as frame 41. Under the forecastle are located the quarters for four commissioned officers, a bath room and toilet, the officers' pantry, the ship's galley, and a crew's watercloset.

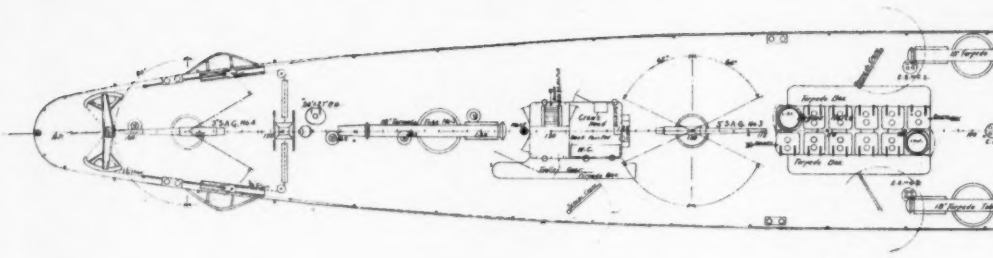
There are four smoke pipes, oval in section, raised hatch coamings for the firerooms, four forced-draft down-takes with concave hoods which can be raised to form deflectors for the wind or can be battened down to keep out the sea or weather. The air ducts are about waist high when closed.

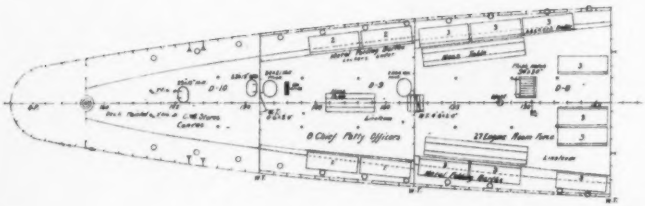
Abaft the smoke pipes on deck is the engine-room hatch, abaft of which is the deck house, including a crew's watercloset and wash room, a wireless-telegraph room and a hatchway to the berth deck.

Under the forecastle forward of the officers' quarters, from frame No. 12, is the lamp room, and forward of that is a small storeroom in the bow. The berth deck extends as far aft as frame 48, and aft from frame 124 to frame 161, or to the rudder post. Between frames 16 and 48 are the petty officers' quarters and the crew's space, separated by a watertight bulkhead with a watertight door, the petty officers' space being

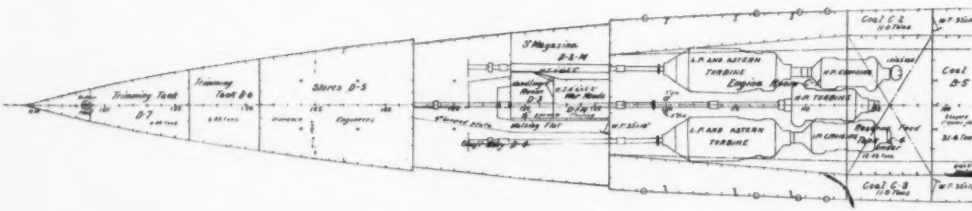
forward of the crew's space. The latter is reached from the main deck by a ladder, and the petty officers' space is reached through the crew's space. Forward of the crew's space on the berth deck are an electrical-supply storeroom and the paint and oil storeroom. A smaller storage space is available in the bow. The narrow space in the bow is filled in with concrete of coke and cement. The two forward storerooms are reached from the deck above, and the electrical storeroom is reached through a door in the forward watertight bulkhead of the petty officers' space. These living spaces are sheathed with No. 27 U. S. G. sheet-iron cork, painted. In the petty officers' quarters are fourteen metal folding berths, and in the crew's space are eighteen of these berths. One mess table is provided for the petty officers' and two for the crew's space. As the crew's space is adjacent the forward boiler space, the bulkhead between is insulated by one inch of asbestos and magnesia, sheathed by No. 27 sheet iron, on the after side. These quarters and storerooms are lighted by dead lights and air ports in the ship's sides. There is no artificial ventilation excepting electric fans in living spaces.

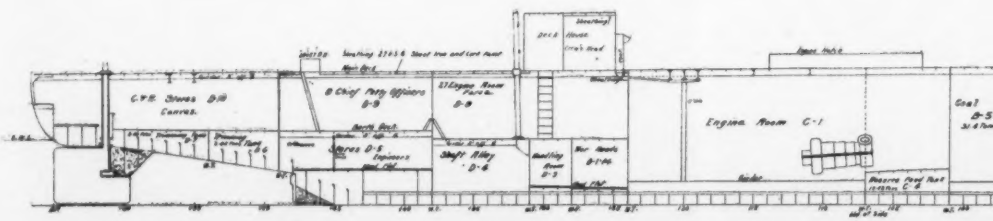
The after berth deck is divided as follows: From frame 124 to 138 are the quarters for 27 men of the engineer force. From frame 138 to 149 are quarters for 8 chief petty officers. Metal folding berths for the engine-room force are arranged three tiers high, and for the chief petty officers two tiers high. There is one mess table in the latter and two in the former compartments. Abaft of the chief petty officers to the stern is a C. and R. storeroom. Under the engineer-crew space are a magazine for war heads, a handling room and the shaft alley. Under the chief petty officers' quarters are the engineers' and ordnance storerooms. Under the C. and R. storeroom are two trimming tanks of the following capacities: D–6, 6.85 tons; D–7, 4.40 tons. From frame 148 to the after perpendicular the stern is cut away, sloping up to the water line excepting for a fin extending four frames from the rudder post forward. The living spaces and storerooms on the berth deck aft, like those forward, are provided with air ports and dead lights.



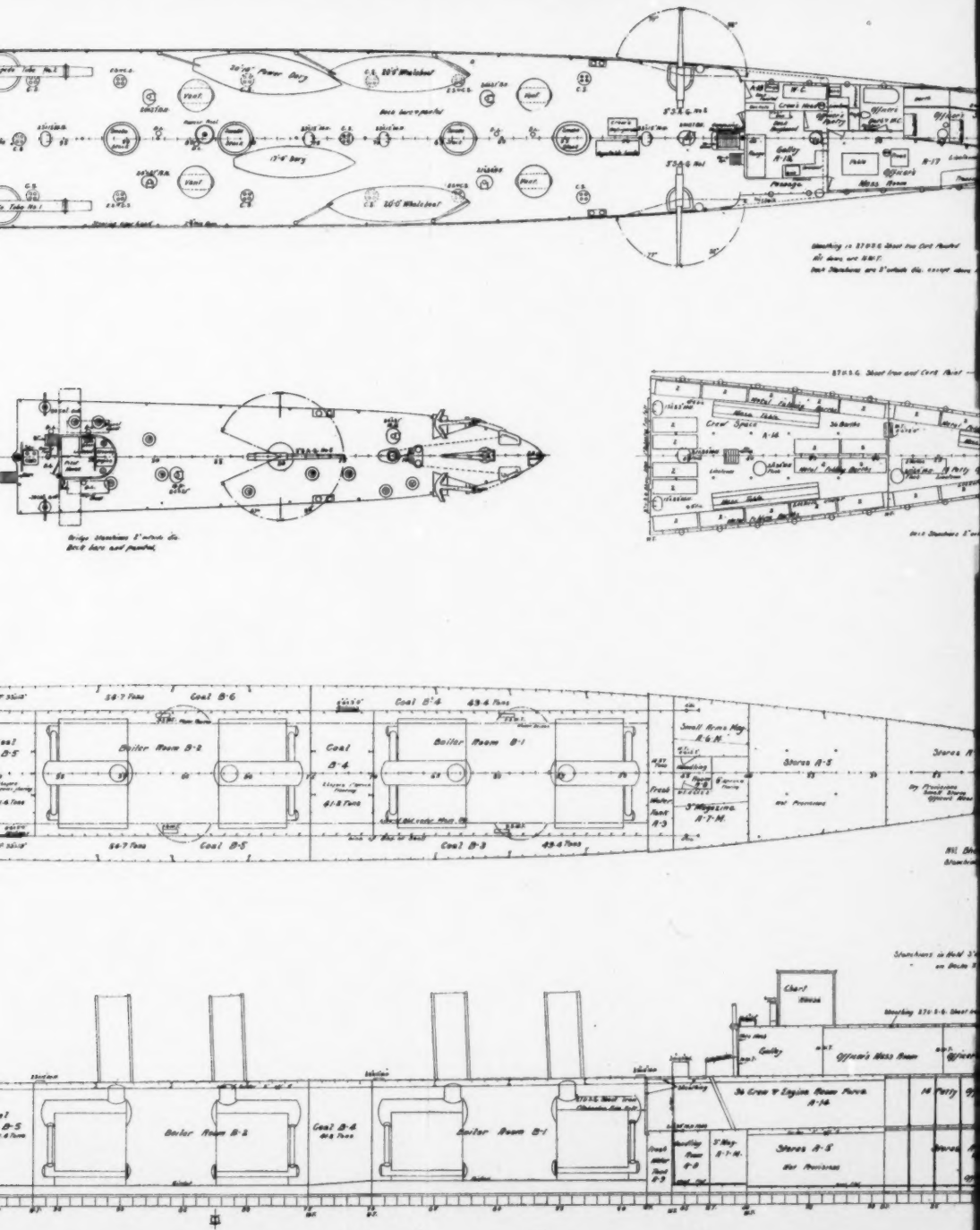


Deck Stanchions 5' On Centre thro. except where marked
Sheathing 2 T.G.S.G. Steel Iron and Cork Floor

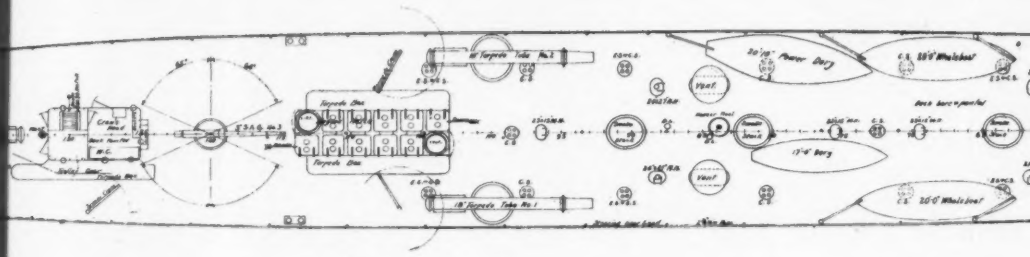


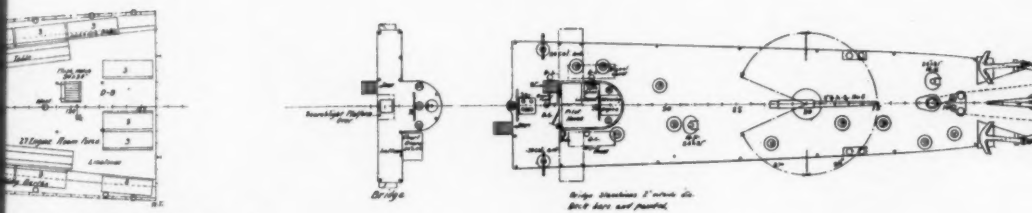


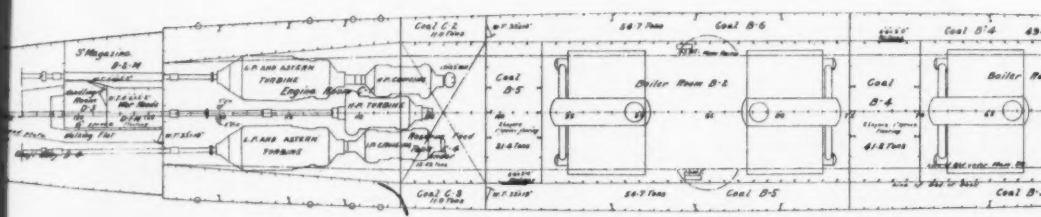
DECK PLANS AND INB

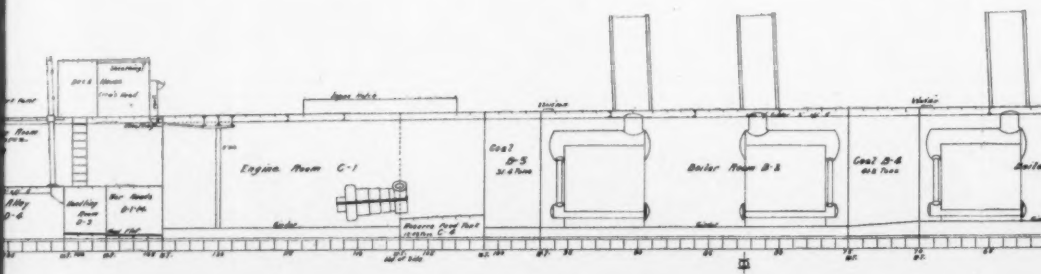


NBOARD PROFILE, DESTROYERS *FLUSSER* AND *REID.*

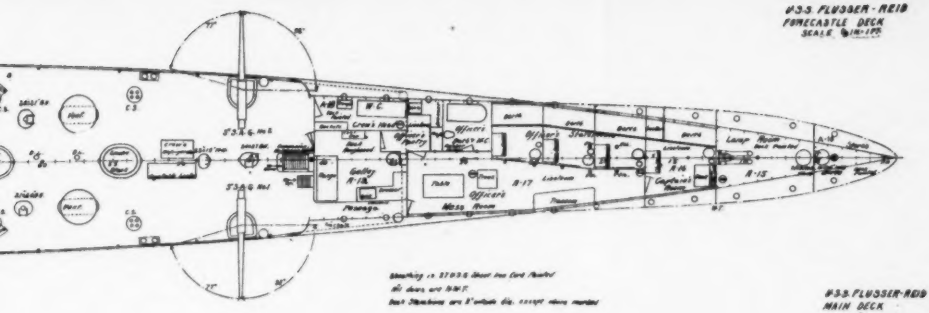






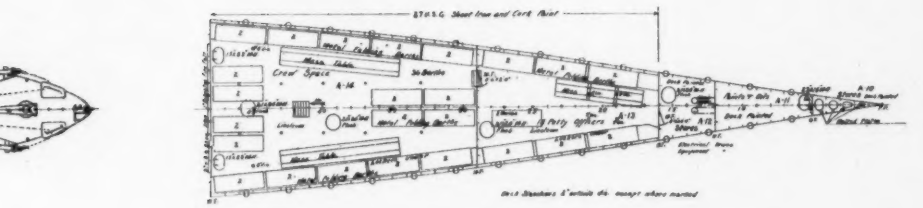


DECK PLANS AND INBOARD PROFILE, DESTROYERS *FLU*

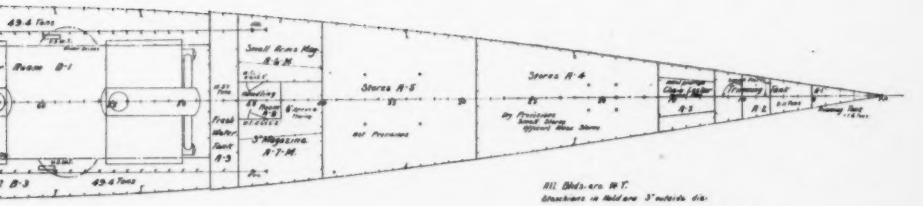


F.S.S. FLUSSER-REID
FORECASTLE DECK
SCALE 1/16"=1'0"

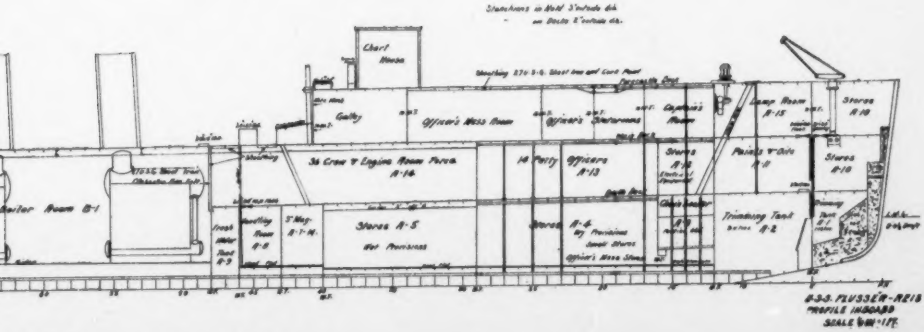
F.S.S. FLUSSER-REID
MAIN DECK
SCALE 1/16"=1'0"



F.S.S. FLUSSER-REID
BERTH DECK
SCALE 1/16"=1'0"



F.S.S. FLUSSER-REID
HOLD
SCALE 1/16"=1'0"



F.S.S. FLUSSER-REID
PROFILE INBOARD
SCALE 1/16"=1'0"

LUSSER AND *REID.*

Under the crew's space forward, from frame 40 aft, are a fresh-water tank, a handling room and 3-inch magazine, and wet-provision storeroom. Under the petty officers' space are the dry provisions, officers' mess stores and small storeroom. Next comes the chain locker (under the electrical storeroom) and next (under the paint and oil room) is a 9.11-ton trimming tank, forward of which, in the fore foot, is a one-ton trimming tank.

There are two boiler compartments and one engine compartment. In the engine-room compartment is a reserve-feed tank of 12.42 tons' capacity, between frames 101 and 107.

The capstan is located on the forecastle with its engine secured immediately underneath on the watertight bulkhead between the captain's and the lamp rooms. It is a diminutive vertical-cylinder (two) steam engine, built by the Hyde Windlass Co., and is controlled by a lever from deck. On the trials this engine hove in 59 fathoms of chain and the anchor in 8 minutes 33 seconds, or at the rate of 6.9 fathoms a minute.

The machinery spaces extend from bulkhead at frame 40 to frame 123, exclusive of the shaft alley, which latter extends as far aft as frame 138. The engine room is included between frames 107 and 124.

Coal bunkers extend alongside the boiler spaces and abreast the make-up feed tank to frame 107 of the engine-room space. In addition to these fore-and-aft bunkers there are two athwartship bunkers, one between the boiler compartments and one between the after boiler compartment and the engine room.

The following are the particulars of the hull:

Length between perpendiculars, feet and inches,	289–00
Projection forward of F.P., feet and inches,	2–02
aft of A.P., feet and inches,	2–08½
Length over all, feet and inches,	293–10½
on load water line, feet and inches,	289–00
of straight keel, feet and inches,	240–07½

Breadth, molded, feet and inches,	26–05
extreme, over fender,	27–01½
Depth, molded, main deck side at frame 83, feet and inches,	16–04⅝
Draught, normal, feet and inches,	8–00½
Displacement, normal, tons,	700
Displacement per inch at designed draught of 8 feet ½ inch, tons,	11.9
Ratio of length to beam,	10.9
Area of immersed midship section, square feet,	135
L.W.L., plane, square feet,	5,015
wetted surface, square feet,	7,150
C. G. of L.W.L. plane aft of M. S., feet,	2
C. B. above bottom of keel, feet,	5.2
aft of M. S., feet,	0.2
Transverse metacenter above C. B., feet,	8.2
Longitudinal metacenter above C. B., feet,	830
Coefficient of fineness, block,	0.405
midship section,	0.644
L.W.L.,	0.666
cylindrical,	0.627
Rudder axis fwd. of after perpendicular, ft. and ins.,	6–06
Strut center fwd. of A.P. center shaft; ft. and ins.,	16–01½
wing shafts, ft. and ins.,	23–01½
Number of frames, spaced 21 inches,	164
watertight compartments,	34

COMPLEMENT.

1 Commanding officer,		
3 Officers,		
85 Crew,	Seamen branch,	27
	Engine-room branch,	51
	Special branch,	2
	Commissary branch,	2
	Messmen branch,	3

CAPACITIES OF TANKS.

Compartment.	Gallons.	Tons F.W.	Tons S.W.
Gravity tank,	50	....	....
Reserve feed tank,	3,335	12.42	....
Trimming tank A–1.	270	1.0	1.03
A–2,	2,445	9.11	9.34
D–6,	1,840	6.85	7.03
D–7,	1,180	4.40	4.51

COAL BUNKERS.

Capacity to 6 inches below deck beams 43 cu. feet = 1 ton.

Compartment.	Cu. feet.	Tons.	Compartment.	Cu. feet.	Tons.
B–3	2,130	49.5	B–6	2,365	55.0
B–4 side.	2,130	49.5	B–5 center.	1,365	31.7
B–4 center.	1,775	41.3	C–2	473	11.0
B–5 side.	2,365	55.0	C–3	473	11.0
			Total,		304.0

HEIGHTS.

Above designed L.W.L., which is 8 feet ½ inch above molded base line.

Top of masts, fore and main, ... 60 feet.
Searchlight platform, ... 31 feet 1¼ inches.
Bridge at center, ... 24 feet 4¼ inches.
side, ... 24 feet.
Top of forecastle deck at stem, . . . 17 feet 3¾ inches.
main deck at stern, ... 8 feet 3½ inches.

Admission and egress to and from the boiler spaces is by raised hatches, one over each side of fireroom and one over the rear end of each boiler. The former have spring catches and glass dead lights, the latter are secured by bolts and are without lights.

MACHINERY.

The motive machinery of the class of destroyers to which the *Flusser* and the *Reid* belong consists of small-tube boilers of the express type and steam turbines, with the usual auxil-

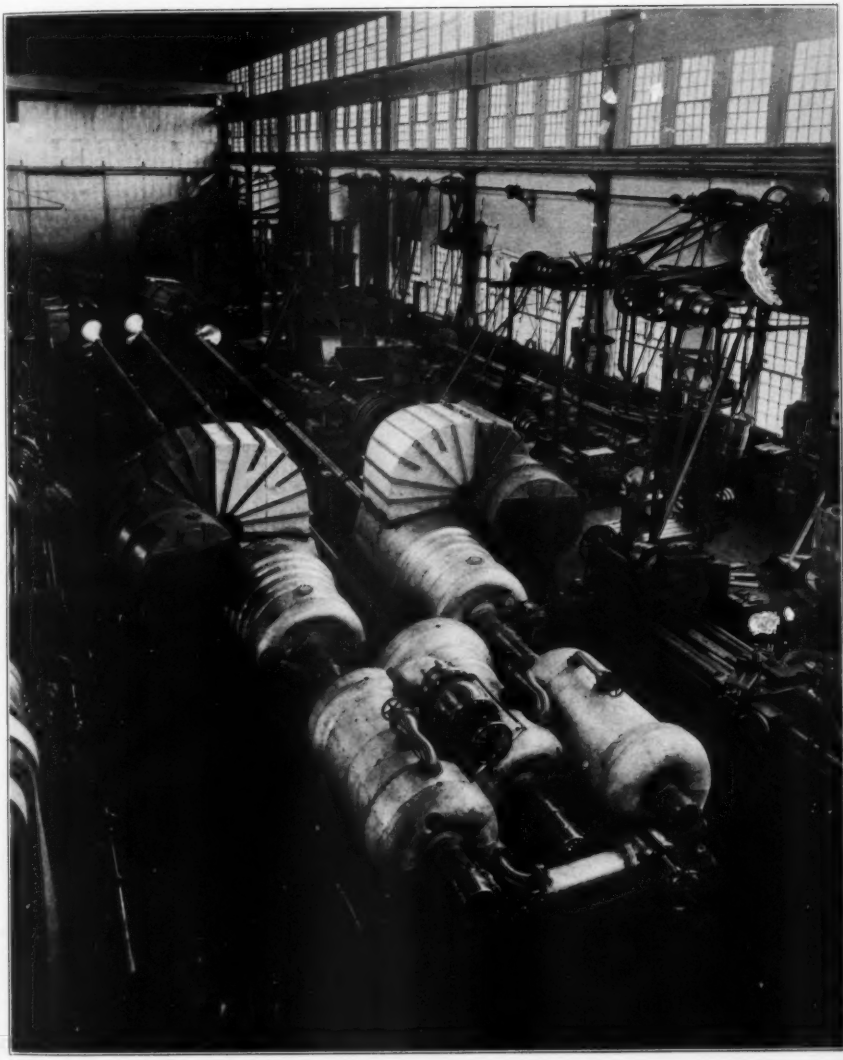
iary machinery. Those having Parsons turbines are *Smith*, *Lamson*, *Preston*, *Flusser*, *Reid*, *Paulding*, *Drayton*, *Roe*, *Terry*, *McCall*, *Burrows*, *Monagan*, *Trippe*, *Ammen* and *Patterson*. Those having Curtis turbines are the *Perkins*, *Skerritt* and *Walker*. Zoëlly turbines are being installed in the *Warrington* and *Mayrant*.

Although the general features of the Parsons turbines are commonly understood and familiar to most engineers of studious habits, there are yet many details peculiar to each installation of interest to many readers of our journal.

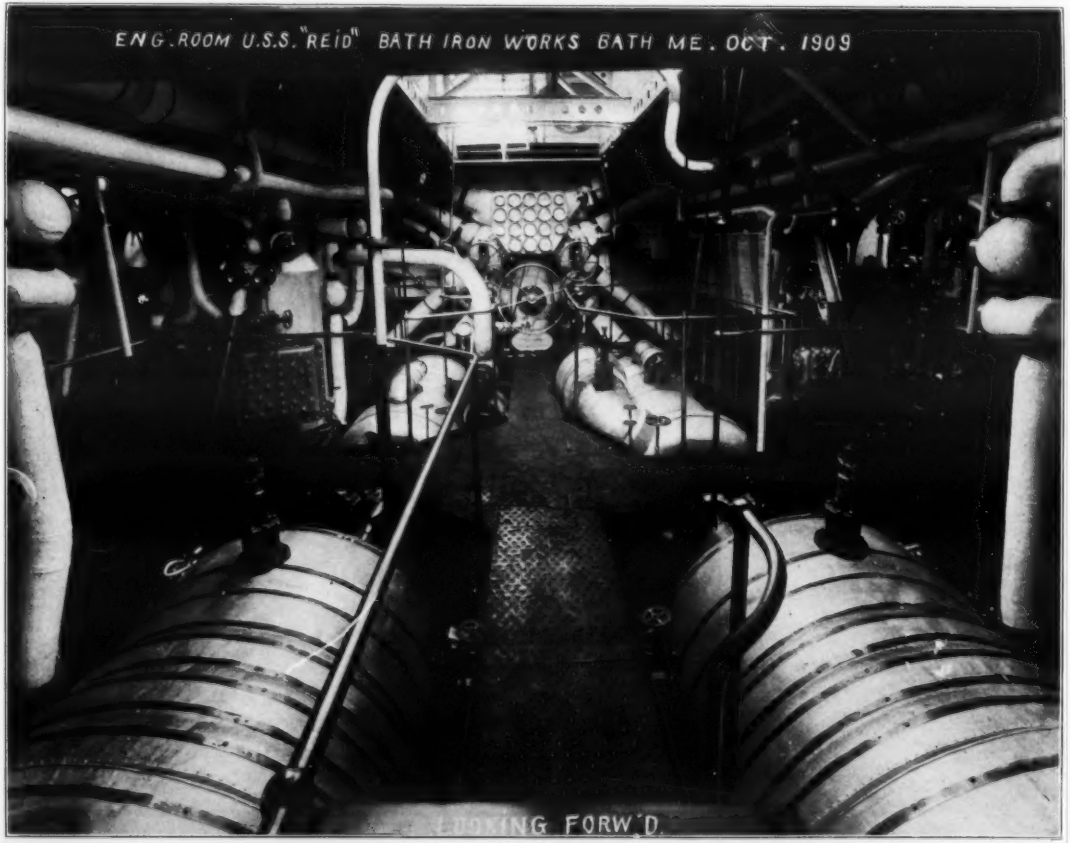
The *Flusser* and *Reid* are our pioneer destroyers—making over thirty knots on their official trials—to be delivered to the Government. By reference to the several views illustrating this article showing the machinery of these vessels, both in the shop and in the ships, it will be seen that there are three shafts driven by five turbines. The middle shaft is driven by the main H.P. turbine; each wing shaft by two turbines for the ahead motion and by one for backing. On the port shaft are the H.P. cruising, the L.P. and astern turbines—the latter two in one casing, the backing turbine aftermost. On the starboard shaft are the I.P.C., the starboard L.P. and astern or backing turbines. Both backing turbines are in the casings of the L.P. turbines.

The working platform is at the forward end of the engine room, and the numerous pipes and valves included in the working gear are pendant from the forward watertight bulkhead of the engine room, which has been stiffened on its forward side for this purpose.

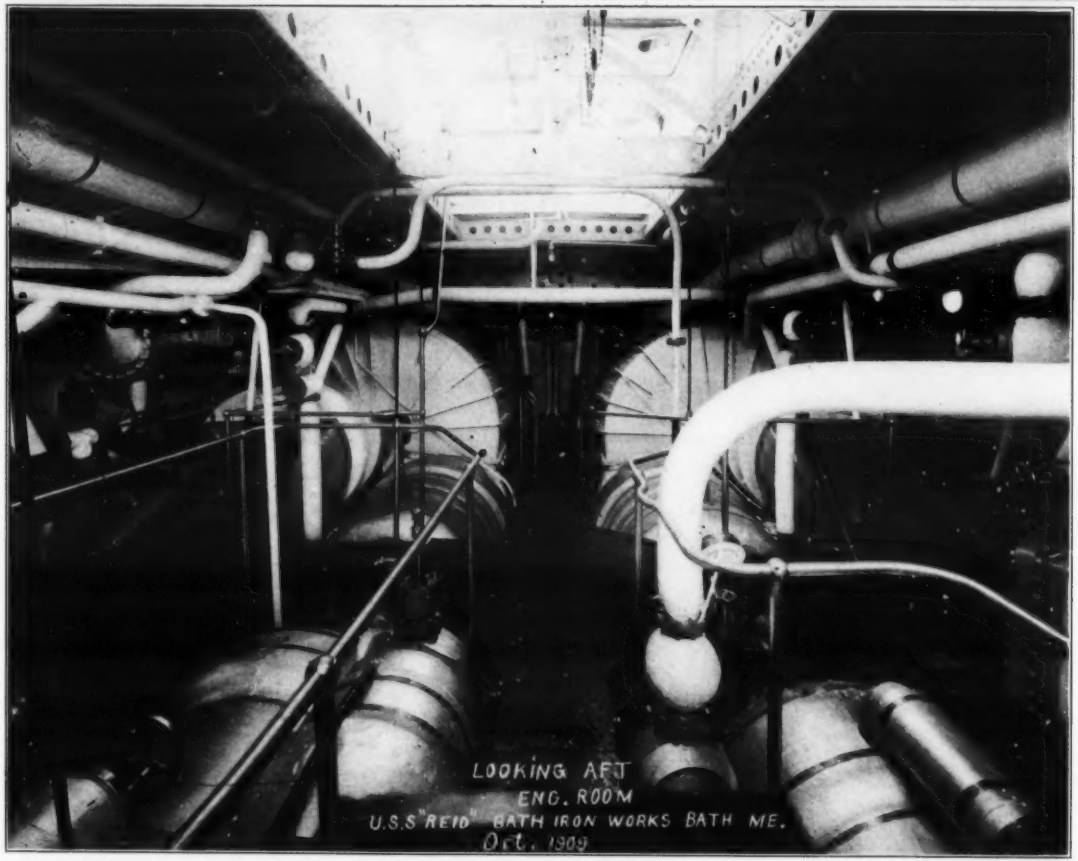
Composing the group of valves at the working platform are two 9-inch, angle, main stop valves, one on each side; the port valve admitting steam from the two after boilers, the starboard valve admitting steam from the two forward boilers. By converging pipes from these bulkhead stop valves the boiler steam is led into a main governor valve, located amidships and uppermost of the distributing group of valves, from which it passes into a tee immediately underneath, the lower (12-inch) nozzle of which is connected with the main H.P.



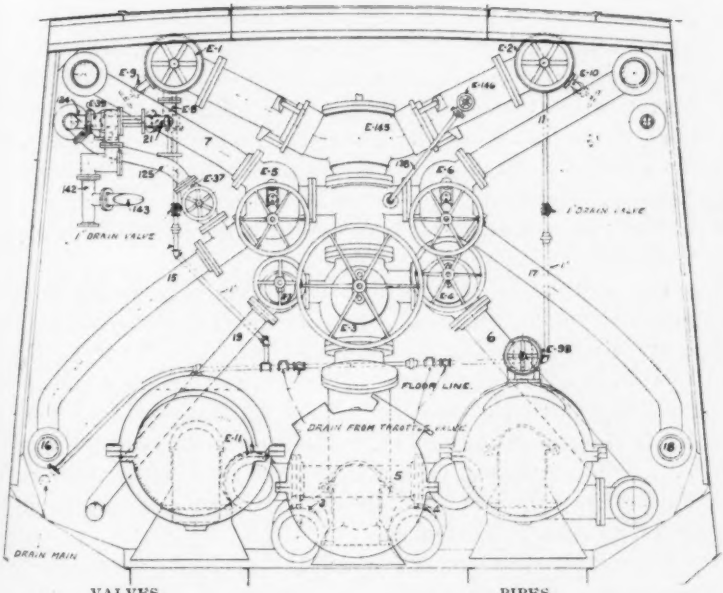
TURBINES, SET UP IN SHOP.



Engine Room, "Reid," Looking Forward.



ENGINE ROOM, "REID," LOOKING AFT.



VALVES.		PIPES.	
E-1, E-2	Main bulkhead stops.	6	Steam to I.P. cruising turbine.
E-3	Main throttle valve.	7-11	" " astern turbines.
E-4	Throttle, I.P. cruising.	15-17	" " L.P. ahead (maneuvering).
E-5, E-6	Maneuvering throttle.	19	" " H.P. cruising turbine.
E-7	Throttle, H.P. cruising.	125	Aux. exhaust to maneuvering steam.
E-8	Bleeder valve.	142	" " " gland piping.
		143	" " " main H.P. turbine.

Fig. 1.—ARRANGEMENT OF VALVES AT FORWARD BULKHEAD OF ENGINE ROOM.

throttle-valve chamber. The latter debouches into a steam strainer immediately underneath, which latter has two nozzles, both leading to the M.H.P. turbine.

On the horizontal arms of the tee aforesaid are two combination maneuvering valves for admitting and regulating a direct flow of steam to the L.P. and astern turbines on each side of the ship. These maneuvering valves are of the double-beat variety, with their spindles connected by a species of "walking beam" fitted with stiff springs over each valve stem and so arranged that in the process of opening one the other valve is closed. The upper combination valve leads to the

ENGINE ROOM, "REID," LOOKING AFT.

U.S.S. "REID" BATH IRON WORKS BATH ME. Oct., 1909

astern turbine on its side; the lower to the L.P. (ahead) turbine on the same side. By turning the maneuvering-valve wheel right handed the lower valve is closed, and by the same operation the upper one is opened; in other words, steam is shut off the L.P. turbine and admitted to the astern turbine. To maneuver ahead, turn the maneuvering-valve wheel left handed; to go astern, turn the maneuvering-valve wheel right handed; or turn in the direction of the hands of a clock when wishing to go astern.

On each side of the M.H.P. throttle-valve chamber is a nozzle to which is secured a throttle valve for admitting and controlling steam for the cruising turbines. The H.P.C. throttle is on the port side and connects through a 5-inch steel pipe with the after end of the H.P.C. turbine. The I.P.C. throttle on the starboard side is connected with its turbine by a 7-inch steel pipe at the forward end of the turbine. Both these valves are double-beat valves, operated by hand wheel on each valve stem.

On the pipe connection to the P.L.P. turbine from the port maneuvering-valve nozzle is a tee and valve connection with the auxiliary exhaust pipe.

Combinations of five, four and three turbines are permissible, either by disconnecting the cruising turbines or by permitting them to revolve in vacuum. For cruising at low speeds the H.P.C. is made the initial turbine by using its throttle as the modulator. The steam enters the after end of the H.P.C., exhausts across to the forward end of the I.P.C. turbine (through non-return valves), which exhausts in turn into the M.H.P. turbine (through non-return valves), which latter turbine exhausts under all conditions into the two L.P. turbines.

At higher moderate speeds the H.P.C. turbine is put out of use, and at the highest speeds the M.H.P. turbine is made the initial turbine.

Some discussion as to the utility of the H.P.C. turbine has resulted abroad with a rather general opinion that it could be

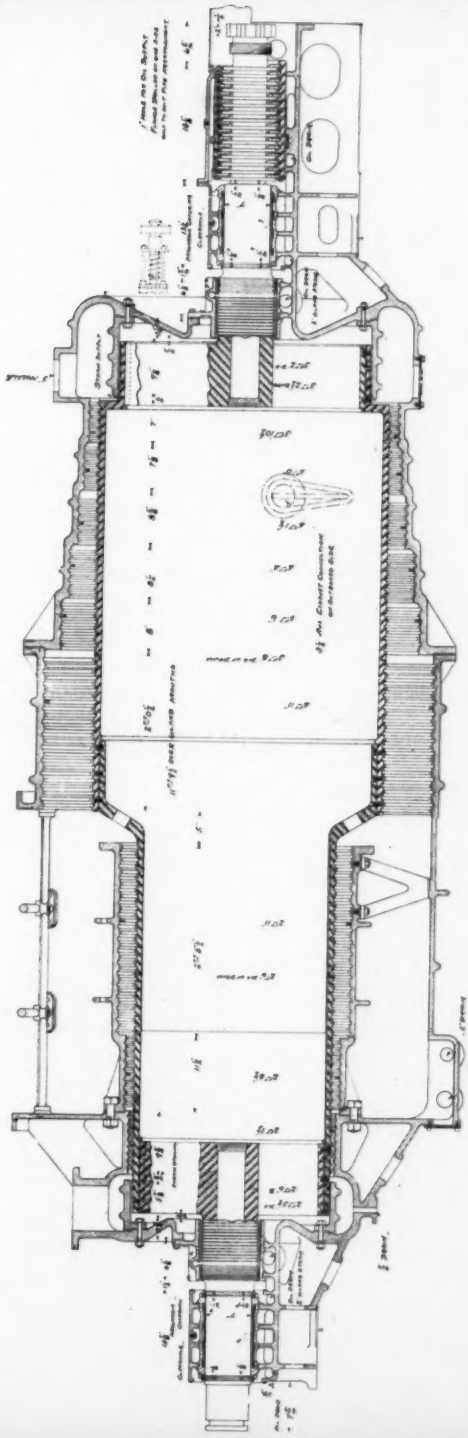
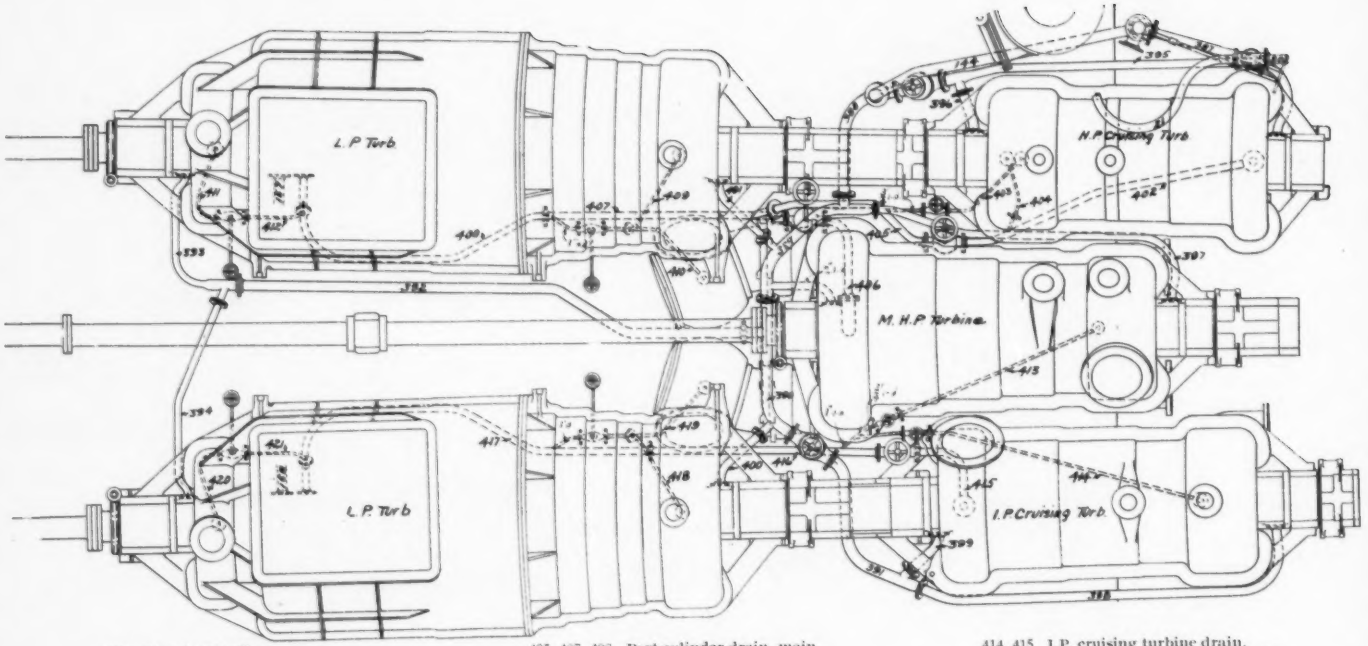


Fig. 2.—Section of L.P. and Astern Turbine, "Flusser" and "Reid."



387 Gland leak off.
144 Auxiliary exhaust to gland piping.
388–401 Gland connecting piping.
402–404 H.P. cruising turbine drains.
405, 407, 408 Port cylinder drain, main.
406, 413 Main H.P. turbine drain.
409 Port L.P. turbine drain.
411, 412 Port astern turbine drain.
414, 415 I.P. cruising turbine drain.
416, 417 Starboard cylinder drain, main.
418 Starboard L.P. turbine drain.
420, 421 Starboard astern turbine drain.

Fig. 3.—Diagram Arrangement of Turbine Cylinder and Gland Drain Pipes.

spared without any considerable economic loss and with a saving in space and weight.

From the L.P. turbines the exhaust is led through large nozzles of rectangular section connecting the top of each L.P. turbine with its adjacent condenser.

There are two condensers of the Normand type, located one on each side of the engine room, elevated considerably above the turbines, resting in three saddles well up against the ship's side. They are built up of steel plates in three sections with brass-tube plates at their ends. Over each end, as collectors and hoods, is a plate-brass cone connecting each end of the condenser with an outboard valve. The circulating water is taken up through a scoop on the ship's bilge, and after passing through the tubes, descends and makes its exit through a similar outlet at the after end of the engine-room bilge.* The tubes are of brass of the following composition: 70 copper, 1 tin and 29 zinc, and are expanded into the tube sheets, which latter are 1½-inch thickness. To allow for expansion of the tubes the condenser and tubes are bent in horizontal planes, to a radius of twenty-three feet. As the tubes are fixed at each end, as they expand they move laterally. They are supported at three intermediate points of their lengths by bridges at the bottom of the condenser. At each point of support over these bridges, and between the layers of the tubes, are placed strips of brass one-eighth of an inch thick, three-quarters of an inch wide, and turned up at their ends where they fit into grooves formed by channel steel riveted to the interior sides of the shell. As the tubes are staggered, and there are no other supporting plates, these strips are necessary to prevent sagging of the tubes. During the shop testing for tightness of these condensers one tube of a lower row was renewed without difficulty by the employment of a solid-pointed head temporarily placed in the condenser tube to be installed.

406, 413 Main H.P. turbine drain. 418 Starboard L.P. turbine drain.
144 Auxiliary exhaust to gland piping. 409 Port L.P. turbine drain. 420, 421 Starboard astern turbine drain.
388–401 Gland connecting piping. 411, 412 Port astern turbine drain.
402–404 H.P. cruising turbine drains.

Fig. 3.—Diagram Arrangement of Turbine Cylinder and Gland Drain Pipes.

* Both intake and outlet are fitted with scoops, so that in backing the normal outlet serves as an injection orifice.

PARTICULARS OF THE CONDENSERS (EACH).

Length of shell over tube heads, feet and inches	11–6¼
Outside diameter of shell, feet and inches	4-5
Thickness of tube sheets, inches	01½
shell plates, inch	00½
Number of tubes, each condenser	2,155
Outside diameter of tubes, inch	00⅝
Thickness of tubes	No. 18, B.W.G.
Length of tubes, longest, feet and inches	12–7¼
shortest, feet and inches	10–9⅞
Cooling surface, outside surface of tubes, square feet	4,003
Area through tubes, square inches	470
of circulating water inlet and outlet, square inches	240.53
through tubes divided by circ. area of inlet, square inches	1.95

There are the following connections to each condenser:

One main exhaust, rectangular, area, square feet	13
auxiliary exhaust, diameter, inches	03½
bleeder (one condenser only), diameter, inches	02½
feed-heater drain, diameter, inches	02½
Evaporator steam-head drain on port condenser	
Heater drains	
One boiling-out connection, diameter, inch	00¾
safety valve, diameter, inches	02
air-pump suction, diameter, inches	10
soda-tank connection, diameter, inch	01
feed make-up (one condenser only), diameter, inch	00¾

The following connections are on each water chest:

One main injection on forward chest, diameter, inches	17½
outboard delivery on after chest, diameter, inches	17½
circulating-pump connection on forward chest, diameter, inches	07
drain, diameter, inch	00½
drain, diameter, inch	00½
air cock at highest part of water chests (piped), diameter, inch	00¼
thermometer connection	
manhole and cover, diameter, inches	12

The condensers are stiffened by five encircling bands of bulb angles 3 inches deep, and the shell is built up of three cylindrical sections with riveted seams, chipped and calked, the axes forming a broken line of three segments.

Auxiliary circulating pumps, one to each condenser, are

of the centrifugal type, driven by a single-cylinder, 3 inches by 3 inches, direct-acting engine, with piston valve, vertical, using steam of boiler pressure, are installed near the forward end of each condenser. On their suction pipes are interlocking sea and bilge suction valves. A trial of one of these engines and pumps, made on a floating derrick with steam from boiler A, for the *Flusser*, gave a maximum capacity of 1,230 gallons per minute. The specifications for these circulating pumps required a collective discharge of two thousand gallons of water a minute. The diameter of the runner of each is fourteen and three-eighths ($14\frac{3}{8}$) inches.

The air pumps, of which there are two, of the duplex-air, single, steam-cylinder, Blake make, are located as low in the vessel as possible, in the after end of the engine room close to the after bulkhead, their working beams in fore-and-aft planes equally distant from the center line of the ship. (For dimensions see list of pumps.) To examine the valves of these air pumps will not be as easy as in large ships, because of the multiplicity of pipes in their vicinity made necessary by the installation of a Parsons vacuum augmentor.

The augmentor consists of a steam ejector, which takes its suction from the tops of the large suction pipes, between the air pumps and the condensers, and discharges into a small condenser named the augmentor condenser, and through it into a water-seal box in such a way as to force the discharge from it into the channel ways of the main air pumps. By the use of this augmentor the vacuum was increased an inch and a half below that possible without its use, and to the excellent vacuum obtained by its use is attributed, in large degree, the high speed and economic performance of these vessels.

On page 520, Vol. XXI, of the JOURNAL, in an article by Lieutenant A. F. H. Yates, U. S. N., is shown an arrangement of the augmentor as applied to the machinery of the *Chester*. Saving for the connection of the augmentor with the down-take pipes from the main condensers to the air pumps, this sketch will suit the *Flusser*.

PARTICULARS OF THE AUGMENTOR CONDENSER.

Material of the shell, copper, formed to a curved cylinder slightly enlarged at its middle; thickness of shell, inch	.1375
Tube sheets and manifold heads are of composition.	
Thickness of tube sheets, inch	00$\frac{7}{8}$
outer heads, inch	00$\frac{1}{2}$
Radius of inner or shortest element of shell, feet and inches	6–03$\frac{1}{2}$
outer or longest element of shell, feet and inches	4–05$\frac{9}{16}$
all tubes, feet and inches,	5–05$\frac{1}{2}$
Versine of angle of planes of the two tube heads to one another, multiplied by radius of all tubes (curvature of), inches	14$\frac{5}{8}$
Number of tubes	225
Diameter (outside) of tubes, inch	00$\frac{5}{8}$
Length of longest tube (by drawing), inches	53
Greatest diameter of shell, inches	21

The tubes are not tinned, and are arranged as described for the main condensers.

As to the additional weight of the augmentor and augmentor condenser, the total for the *Flusser* was as follows:

One vacuum augmentor, pounds	145
augmentor condenser, pounds	745
Steam and water piping and valves, pounds	1,143
Total	2,033

The circulating water is taken from a branch on the inlet nozzle of the port condenser and discharges through a branch on the outlet nozzle from the starboard condenser. Each of these branches is fitted with a scoop to insure a circulation of water while the ship is backing as when going ahead. The augmentor and the air-pump connections are at the same end, top and bottom of the shell, respectively, a baffle plate being fitted under the bottom row of tubes, with an opening at one end opposite from said connections.

THE BOILERS.

There are four boilers of the Normand, return-flame, express type, each with a single furnace and three furnace doors —two boilers in each of the two boiler compartments. To compensate for the squatting of the stern of the ship at high

speeds the boilers are pitched forward, or slightly depressed at their forward ends. This is the Normand practice. For draining the lower drums drain cocks are installed at the forward ends of boilers A and C, the bottom-blow valves on the front ends of the two other boilers serving this purpose. This design of boiler has demonstrated a very high potentiality. On the full-speed four-hour trial of the *Flusser*, with five inches air pressure in firerooms, 34.1 B.H.P. for each square foot of grate surface was developed. At times during the builders' trials over eight inches air pressure was indicated on the air gauges. With five inches air pressure the coal burned per square foot of grate was, for the *Flusser*, 63.9 pounds (Pocahontas).

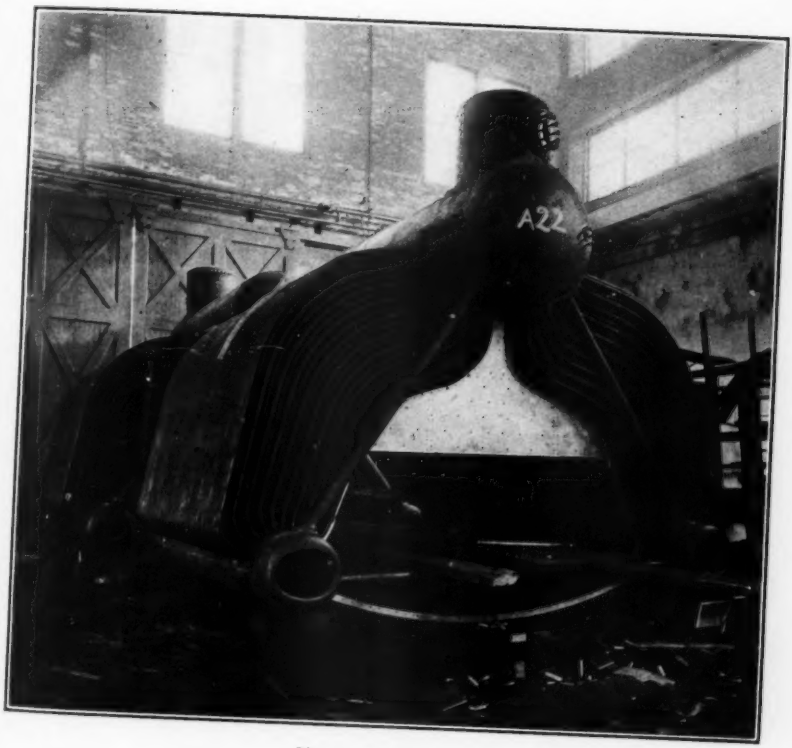
Another uncommon feature of this boiler is the length of grate—8 feet $8\frac{3}{16}$ inches. The grate, is composed of class C, medium-steel plate, lightened by holes and built up in form of double grate bars, their ends resting on the usual bearing bars, which latter are supported by struts at their middle points in addition to end supports.

The interior, or furnace, tubes are placed tangent and parallel to each other and, arching over the furnace, form a very high and spacious combustion chamber, the walls of which, excepting for a short distance from the back end of the furnace, are without opening for the passage of gases of combustion; these gases passing through and among the wider-spaced tubes at the rear of the furnace, returning to the front end of the boiler through sinuous spaces between the tubes, which are staggered for this purpose. The outside row is so installed as to form an impervious wall to the gases, and asbestos cord pressed into the spandrels between the tubes completes the joints. About a third of the way from the front of the boiler is placed a steel athwartship baffle plate to direct the flame from the upper portions of the tubes to the lower, which is to say, to prevent the flame from passing horizontally from the rear to the front, and so on up the uptake without passing among the tubes at their lower portions. This baffle plate is perforated.

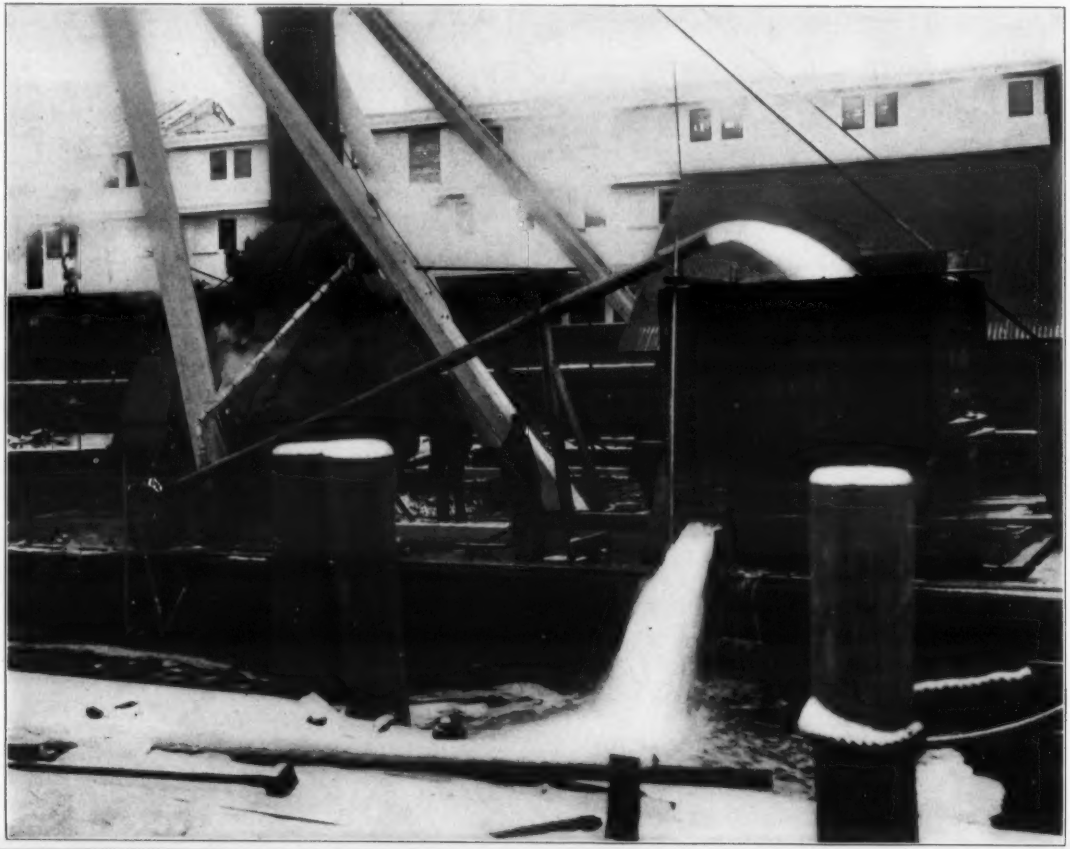
In the *Flusser* and *Reid* the fire-brick linings of the furnace, slightly above the grate level, as usual, are of special form and size. In the later destroyers these bricks are of standard dimensions. All are secured by hook bolts to the tubes in their vicinity, and where there are no tubes near enough for this, as at the front and back of the furnace, the brick lining is secured to the boiler casing, from which it is separated and insulated by air spaces and magnesia-asbestos clothing.

A steam dome near the front end of the steam drum, under which is a baffle plate, insures very dry steam under all conditions of steaming. The dome is partially enveloped by the uptake, but protected from the hot gases by an air space and the wall of the uptake and its included air space and insulation. The height of the dome above the top of the drum is 26¼ inches. The uptake is insulated by a half-inch air space, ¼-inch asbestos board, 1-inch block magnesium, sheathed over all by a No. 22 U. S. S. G. galvanized-steel sheathing plate. There is a twin 4½-inch safety valve on each drum, and the usual appendages. The normal water line is 8 inches below the center line of drum.

If parallel planes were passed through the axes of the upper and lower drums they would be seven feet nine inches apart. The upper drum is composed of two plates, two bumped heads and the steam dome. The lower drums are seamless-drawn steel tubes, reduced in thickness at the portions not pierced for the tubes. The after ends of the lower drums are bumped and riveted in, and have each a large hand hole and plate for examination and cleaning purposes; while at the forward end is a flange riveted on and a cover (convex) bolted to the flange. There are two down-comer pipes at the after end of the boiler, extending from the upper to the lower drums. They are attached by flanged sockets riveted on. At the front of the boiler are two similar but smaller hollow stays between the upper and lower drums and one solid stay between the lower drums at each end.



NORMAND BOILER.



PARTICULARS AND DIMENSIONS OF THE BOILERS (EACH).

Number of boilers (two in each boiler compartment)	4
Designed working pressure, pounds	255
Hydrostatic test applied, pounds	450
Steam-pressure test applied, pounds	300
Grates, length, feet and inches	8-08 5/16
width, feet	10
Per cent. of air space in grates	59 3/8
Grate area, each, square feet	86.82
Number of furnaces	1
furnace doors	4
Heating surface, area in square feet (one boiler)	4,044.25
Ratio of grate to heating surface	1 to 46.5
External height of boilers, feet and inches	13-09 1/4
width of boilers, feet and inches	15-03
length of boilers, feet and inches	13-03 13/16
Tubes, outside diameter of, inches	1 3/16
thickness of	No. 11, B.W.G.
kind of	Cold-drawn seamless steel.
number to each boiler	1,612
longest, length, feet and inches	10-08 23/32
shortest, feet and inches	7-11 1/32
Smoke pipes, number of	4
height above grates, feet and inches	24-09 3/4
area of section, each, square feet	12.18
Ratio of grate area to area of each smoke-pipe section	7.13 to 1
Drum, upper, diameter of inside, inches	39 3/8
thickness of top sheet, inch	19/32
lower sheet, inches	1 1/8
Dome, steam, diameter of inside, inches	27 9/16
thickness of, inch	9/16
The upper drum heads are bumped to a radius of, inches	39 3/8
have thickness of, inch	19/32
The dome head is bumped to a radius of, inches	27 9/16
has thickness of	9/16
is braced to a stiffening ring of angle steel whose projected diameters are, inches	17 1/2
The aforesaid stiffening ring is riveted to the shell and is made of angle steel, inches	3×4×3/8
Dome braces, diameter of, inch	1
number of	3
Down-comer pipes, diameter of, inside, inches	10
outside, inches	10 1/2
Tie rods, hollow, front, of mild steel (class B), outside diameter, inches	3 1/2
Tie rods, hollow, front, of mild steel (class B), inside diameter, inch	1/2

Lower drums, diameter of inside, inches..... 15½
thickness of top sheet, inch..... 11/16
lower sheet, inch..... ⅝
Horizontal tie rod between lower drums, solid, diameter of (one of these at each end of boiler), inches..... 2½
Weight of one boiler with all appendages complete, 47,434 lbs. = 21 804/2240 tons.
water and steam at steaming level, 250 pounds gauge pressure, 7,102 pounds = 3 472/2240 tons.

Spaces for passage of gases of combustion are provided by omission of tubes, thus forming diagonal alleys through the nest of tubes at each end of the furnace where necessary.

LUBRICATION.

Lubrication of the main turbine, spindle and line-shaft bearings, including thrust bearings, and of the forced-draft blower engines is accomplished by oil forced through them by two pumps located in the engine room. The oil passes through a cooler on its way to the bearings and is collected from the latter by a system of drain pipes which lead into a tank of one hundred and fifty gallons capacity, located amidships on the starboard side of the inner keel, just abaft the main high-pressure turbine. This tank is so low in the ship as to insure a rapid flow, visible through an oblong pane of glass in the top of a manifold collector at the tank. At the discharge nozzle of each pipe into this manifold is a thermometer. These thermometers are together in an athwartship row, and inclined so that they may be read by the light of an electric lamp conveniently placed, and by an observer standing on the engine-room platform. During the trials a series of observations was made by an observer at close hand, and the results recorded on a bulletin board with chalk, so as to be seen more readily from the working platform. In the circulating-oil line is a cooler located overhead the starboard side of the engine room. Circulating water is from a pipe connection with the auxiliary-pump discharge pipe to the starboard-condenser water chest, and from an emergency connection with the engine-room fire and bilge pump.

The oil cooler is of the surface-condenser type, the tubes of

which are expanded and sweated into the two tube sheets. By baffles on the heads the oil is made to pass through the tubes in six passes. The nests of tubes are triangular in section and include forty-seven tubes each, making a total of two hundred and eighty-two standard tubes of brass, five-eighths of an inch outside diameter. The shell of the cooler is a brass tube flanged at each end. The oil heads are of cast composition, and, like the tubes, is of the following mixture: Copper, 70 per cent.; tin, 1 per cent.; zinc, 29 per cent. By two stiffening rings of class B steel and through bolts and collar bolts the shell tube heads and water heads are bolted together. These bolts are of class B steel.

PARTICULARS OF THE OIL COOLER.

Diameter of shell, inside, inches	17½
Thickness of shell, inch	00⅝
Length of shell, feet and inches	4–05
Tube sheets, thickness of, inch	00⅞
diameter of, and of flanges, inches	21¼
Oil heads, diameter of, inches	21½
thickness of, inch	00¼
Circulating-water nozzles on shell, diameter of hole, inches	3
Tubes, number of	282
outside diameter of, inch	00⅝
Area through each nest, square inches	10¼
of cooling surface, square feet	171.7

The circulating water passes through a nozzle on the side of the cooler and, after zigzagging among and across the outside of the tubes by direction of six baffles, through which the tubes pass, it passes out of a similar nozzle at the top of the cooler shell and discharges overboard above the water line on the starboard side. This discharge water is plainly and conspicuously visible in the views of the vessel under way.

The feed and filter tank is located immediately over the air pumps, and is suspended from the main-deck beams and secured to the after bulkhead. It is built of light galvanized sheet steel, stiffened by angles and braced light angle beams between its sides and ends. At the top are two filter com-

partments, so arranged as to cause the discharge from the air pumps to flow over a weir downward through the loofa into a channel, and up another channel and over another weir into a common middle passage to the feed-tank reservoir below. This insures the constant submergence of the filtering material. The capacities of the upper and lower parts are 150 and 350 gallons, respectively. The connections with the first compartment of each filter section are as follows:

Two main air pumps, diameter, inches	9
Filling pipe, diameter, inches	2
Distiller fresh-water connections, diameter, inch	$0\frac{1}{2}$
Vapor pipe, diameter, inches	2
Drain pipes, with valves, diameter, inch	$0\frac{1}{2}$

On the feed tank are the following connections:

Overflow, diameter, inches	5
Vapor, diameter, inches	2
Two feed suctions, diameter, inches	7
Drain plug in bottom, diameter, inches	$1\frac{1}{2}$

Water-gauge glasses.
Thermometer.

THE FEED-WATER HEATERS.

There are two feed-water heaters, one in each fireroom, in which the feed water is heated by the exhaust steam after the manner of a surface condenser or a dead-end distiller. They are of the Normand type and consist each of a shell of thin steel plate to which are bolted two flat tube heads of steel and two spherical water heads of the same metal. Into the tube heads are expanded three hundred and six brass tubes of five-eighths of an inch diameter (outside) and No. 18 B.W.G. thickness. Inside of each tube is placed a retarder of crimped brass ribbon, $\frac{5}{16}$ inch wide, No. 20 B.W.G. thickness, extending the entire length of the tube. The axis of each heater is placed vertically and is secured to the fore-and-aft bunker bulkhead. Secured by tap bolts over and to the upper tube plate is a thin, perforated plate of steel, the holes in which are $\frac{3}{16}$ inch diameter, pitched $\frac{3}{8}$ inch apart, serving as a scattering plate and a binder to the ends of the baffles which protrude

from the tubes. The holes in the scattering plate are chamfered so as to prevent sealing the holes by contact with the ends of the baffles. In the lower head are four spiral coils of copper pipe, forming a continuous lead from a point eighteen inches above the bottom tube head on the interior of the shell through the spiral to an exit on the lower head, for the condensation of vapor and the exit of air, piped to drain from the heater trap. There are twelve flat baffle plates through which the tubes pass, secured to the inside of the shell by three ears each. Each baffle extends an inch and seven-eighths beyond the center line of the heater and are of $\frac{1}{8}$-inch steel plate. The feed water enters the lower head, passes through the tubes and makes its exit through a nozzle on the upper head. The exhaust steam enters the shell through a nozzle near the top of the shell and is drained to a trap through a nozzle on shell close to the bottom tube head. Through the center of the end heads and through a central tube expanded into both tube heads is an axial stay rod with nuts at each end and on the under sides of the end heads. The shell is formed of a single sheet with a longitudinal double-riveted seam, a flange bolted to each end, which flange is reduced in thickness within the area of the bolt circle so as to form an expansion joint between the shell and the tube plate. A small drain hole is provided by a seven-sixteenths hole plugged by a $\frac{1}{4}$-inch pipe plug through the edge of the lower tube plate and up into the shell space, making a right-angle passage. A gauge glass with fittings attached to shell four (4) and thirty (30) inches above lower flange of shell is installed.

PARTICULARS OF THE HEATER (EACH).

Length over all, feet and inches	10–01
of shell between tube heads (inside), feet	8
Diameter of shell (inside), inches	18$\frac{1}{4}$
Thickness of shell (inside), inch	0$\frac{1}{2}$
Tube heads, diameter of, inches	25
thickness of at tubes, inch	1
flange, inch	0$\frac{3}{4}$
End heads, thickness of throughout, inch	0$\frac{15}{16}$

Central stay, diameter of, inches	1¼
length of, feet and inches	10–01
Baffles, shell space, number of	12
thickness of, inch	0⅛
Tubes, outside diameter of, inch	0⅝
length of, feet and inches	10–00⅝
number of	306
material of; composition: admiralty mixture, 10 copper, 29 zinc, 1 tin.	
Nozzle for feed water, diameter, inches	4
Exhaust inlet, diameter of, inches	5
drain to heater, *via* trap, diameter of, inches	2
Spiral air coil, diameter of pipe (copper), inside, inch	0½
thickness of	No. 19 B.W.G.
length of, feet (about)	56
(Note: A valve bolted to the shell is the upper terminus of this pipe.)	
Heating surface, outside of tubes, square feet	400
Area through tubes, square inches	688
retarders fitted, square inches	33.9
Ratio of area through tubes to outlet	2.6 to 1
Actual weight, including air valve, pipe and water-glass fittings, pounds	2,029

Auxiliary Pumps in the Engine Room.—On the starboard side, secured to the after bulkhead of bunker C–3, are two oil-circulating pumps, Blake, simplex, 6 inches by 7 inches by 8 inches. One fire and bilge pump, secured to the side of the ship, 7½ by 5 by 8 inches, duplex, takes from sea after drainage suction, and has a suction-hose nozzle; discharges overboard, to fire main, to oil cooler, and has a hose connection. By a cross connection to port side it may discharge through the distiller.

On the port side of the engine room are the two evaporators and an evaporator feed pump, size 3½ by 4 by 4 inches, simplex, Blake.

The distiller is located overhead the port side of the engine room, and is so small as to escape observation unless searched out. It is composed of a brass seamless shell, flanged, upon which are bolted the tube heads and end cowls or heads, which are spherical and of copper. On the bottom head (the axis of the shell is vertical) are riveted and sweated two nozzles, one leading forward and the other aft, to the ship's tanks and the feed and filter tanks, respectively. On the upper cowl or

head is a nozzle attaching the vapor pipe from the evaporators. The vapor passes through the tubes from top to bottom, while the circulating water zigzags around them from bottom to top, directed by baffles, three in number, through which the tubes pass and which are bolted by ears to the sides of the shell. The circulating-water, entering and exit nozzles are riveted and sweated to the shell. The tubes are expanded and sweated into the tube sheets. The spherical heads are brazed to composition flanges, the tube sheets are thinned to a quarter of an inch thickness at the flanges, and the shell flanges are secured to the others each by four collar studs, with nuts at both ends and twenty-eight through bolts, all of class B steel. The collar bolts permit the spherical heads to be removed without disturbing tube heads, the through bolts dropping out on removal of the nuts from the ends, while the collar bolts remain to hold the tube sheets.

DIMENSIONS OF DISTILLER.

Shell, brass, inside diameter, inches	08
thickness	12, B.W.G.
Outside heads, copper, radius of, inches	05¼
Distance between inside faces of tube sheets, inches	19
Thickness of tube sheets, inch	00⅝
Diameter of tubes, tinned inside and outside, O. D., inch	00⅝
Length of tubes over all, inches	20¼
Outside diameter of flanges and tube sheets, inches	11¼

SCREW PROPELLERS, THREE BLADED, THREE TO EACH SHIP.

These are solid manganese-bronze, true screw of uniform pitch, the starboard screws right handed, the port screws left handed and the center screws are right handed. The propellers were cast at the works of the Hyde Windlass Co., Bath, Me., and were machined at the builders' works. After applying a template to the casting to determine sufficiency of stock the forward face of the hub was faced in a lathe, after which the propeller was placed upon a platen of a vertical boring mill, secured to face plate of a planer, and so geared as to cause rotation of the propeller about its axis proportionately to the downward feed of the planer-tool head. Each blade

face was planed radially from tip to hub with mathematical accuracy. After leaving the planer the faces of the blades were smoothed by grinding, filing and polishing, the hubs were finished and the wheel was balanced on a mandrel by removing surplus metal from the back of the blade requiring such reduction. The forward end of the hub is fitted with a stuffing box, gland and india rubber packing ring. The shaft sleeve fits snugly into the hub for an inch, leaving a space of about $\frac{3}{16}$ of an inch into which was pumped red-lead putty to protect further the steel shaft from possible corrosion from leakage of sea water. Two longitudinal keys secure the shaft from rotation within the propeller, and a composition nut on the end of the shaft, which latter is tapered, secures the propeller to the shaft. The nut is secured from turning by a keeper consisting of a rectangular piece of composition which projects into one of the spanner slots of a nut and is tap-bolted to the hub.

MEASUREMENT OF THE PROPELLERS.

The pitch of the propellers at one hundred and five spots taken at uniform distances apart on each blade measured finally as follows:

.....	Item.	Designed.	Measured.	Weight.
				Lbs.
Wheel "K".....	Diameter.	63 ins.	63.34 ins.	1,070
Average of all blades.	Pitch.	58 ins.	58.02 ins.	
	Hel. area.	2,128.8 s. i.	2,136.24 s. i.	
	Proj. area.	1,869.0 s. i.	1,861.96 s. i.	
	P. ÷ D.	58÷63=.921.	58.02÷63.34=.916.	
Wheel "A".....	Diameter.	63 ins.	63$\frac{3}{4}$ ins.	1,053
Average of all blades.	Pitch.	58 ins.	57.95 ins.	
	Hel. area.	2,128 s. i.	2,152.80 s. i.	
	Proj. area.	1,869 s. i.	1,854.73 s. i.	
	P. ÷ D.	58÷63=.921.	57.95÷63.094=.918.	
Wheel "C".....	Diameter.	Same as above.	62$\frac{7}{8}$ ins.	1,053
Average of all blades.	Pitch.		58.02 ins.	
	Hel. area.		2,146.5 s. i.	
	Proj. area.		1,844.4 s. i.	
	P. ÷ D.		62.875÷63=.923.	

As shafts have at times become so corroded at the juncture of the propeller and sleeve that propellers have dropped off,

the necessity for care to guard against this, both in the original installation and by precautionary examination during service, is emphasized.

An attempt to seal the joint between the shaft sleeves and casings by pure tin was abandoned because of the excessive expansion of the parts and consequent fracture of the joint as soon as it cooled. These joints were made by calking the sleeves at the joint and pumping red lead under them.

CURTIS STEAM TURBINE (DYNAMO SET).

This small turbine is shown on page 823, Vol. XXI, No. 3, of the JOURNAL, published in August, 1909. The rated capacity of the dynamo is 5 kw.; voltage, 125.

PUMP AND PUMP CONNECTIONS.

No.	Pumps.	Size, Inches.	Kind.	Inches.	Suction pipes from—	Inches.	Discharge pipes to—	Location.
2	Main air.....	10×92×15	Vert., independent.	10	Condenser.	9	Feed ta'ks.	Aft. end of eng. room.
2	Auxiliary circulating.	.....	Vert., single, centrifugal.	7 7	Sea..... ..... Bilge.....	7 ...	Condenser. Oil cooler, starbd. Dist., port.	Two in eng. room.
2	Main feed.....	14½×10×12	Vert. piston, simplex.	5 3½	Feed suct. pipe. Reserve feed tanks.	4	Main feed disch'ge.	One in each fireroom.
2	Aux. feed.....	7½×5×8	Vert., piston, duplex.	2 4 3½	Bottom blows..... Feed suct. pipe. Reserve feed tanks.	3½ 2	Aux. feed. Overboard.	One in each boiler r'm.
2	Fireroom fire and bilge.	7½×5×8	Vert., piston, duplex.	4 4 4	Fireroom floors. .. Sea..... ..... C. & R. drain'ge manifolds, forward fireroom only.	2½ 3½ 2 1½	Fire main. Overboard. Ash ejector. Hose connection.	One in each boiler r'm.
1	Engine-room fire and bilge.	7½×5×8	Vert., piston, duplex.	4 4 1¼ 4	Sea..... ..... Bilge, in engine room. Hose connection. C. & R. drain'ge.	2½ 3½ ... 1½	Fire main. Overboard. Distiller. Hose connection. Oil cooler.	One in eng. room.
1	Evap. feed.....	3½×4×4	Vert., piston, single.	...	Sea.. ..... Dist. circulating water disch.	...	Evap'tors.	Near evaporators.
2	Oil.....	6×7×8	Vert., piston, simplex.	...	Bilge oil tank.	...	Oil-press. system.	Near oil cooler.

BUILDERS' TRIALS.

Builders' trials of the machinery of the torpedo boat *Flusser* were made as follows:

First dock trial,
Second dock trial,
First trial in free route, August 12, 1909.
Second trial in free route, August 19, 1909.
Third trial in free route, August 21, 1909.

On August 21 the contractors made several runs over the Southport measured-mile course. The I.P.C. thrust bearing and the main M.H.P. heated and was overhauled and refitted.

On August 30, 1909, at 10:09 A. M. the *Flusser* left Bath, bound for Rockland, for the official preliminary acceptance trials. While steaming with the I.P.C. as the initial, turbine speed had to be reduced at 1:45 P. M. by reducing the pressure in the I.P.C. turbine from 140 to 100 pounds, because of heating of the L.P. thrust bearing on the same shaft. The peculiar condition of excessive forward thrust on this shaft when the initial pressure of 140 pounds was carried in this turbine disappeared when the initial pressure was increased considerably, as, say, to two hundred pounds. To overhaul the thrust of the starboard L.P. turbine the vessel was anchored in Tennants Harbor at 2:15 P. M. At 4:15 P. M. the vessel left for Rockland, arriving there at 5:30 P. M.

On the 31st the boat was taken out for trial and to get the thrusts in good condition by wear. On the following day, September 1st, the standardization trials were made by the Trial Board, over the Rockland course, as described further on.

In reference to the troubles with the thrust bearings of the *Flusser* on the builders' trials, the cause might easily have been attributed to insufficient bearing surface of the thrust bearings. The builders seemed to be of the opinion that the composition of the thrust collars—88, 10 and 2—was not suit-

able for this fast-running shafting (900 revolutions). Therefore a trial bearing was made with collars of a mixture of copper, tin and lead instead of zinc. The behavior of this mixture was certainly an improvement over the Government composition, as the rings did not smear and drag nor heat under the same conditions which caused trouble with latter composition. But I am not prepared to state that the surface of the thrust rings after the trial was all that was to be desired as to smoothness, as to polish and as to wearing qualities. That the *Flusser* and the *Reid* went through their trials so creditably and with a great excess of speed over the requirements is sufficient vindication of the utility of the composition of the emergency or experimental brasses of the lead mixture. Their wearing qualities can be demonstrated later and leisurely during the commission of these vessels. On this subject it is to be hoped that the readers of the JOURNAL will receive some information from the officers of the vessel before the subject is shelved and forgotten.

TORSION METERS.

Shaft horsepower was calculated from the torsional deflection of the shafts, which was observed through use of the "Chadburn's Flashlight Torsionmeter," which is illustrated and described in Volume XXI, No. 2, of the JOURNAL. One of these meters was applied to each of turbine shafts, with the lamp forward, in the engine room abaft the turbine, the finder in each case located in the shaft alley in an athwartship line forward of the spring bearing. To provide a suitable line of sight through the bulkheads holes were cut therein which were afterward patched. Brackets of angle steel secured the lamps and finders rigidly to the hull. As the instruments were loaned by the contractors for use during the official trials, the regular installation for the ship not having arrived in time, all the fittings were not permanently installed for these instruments.

OFFICIAL TRIALS OF "FLUSSER."

STANDARDIZATION TRIAL.

(*a*) The standardization trial of torpedo-boat destroyer *No. 20* (*Flusser*) commenced at 6:38:15 A. M. and ended at 9:40:18.8 A. M., September 1, 1909, over the Rockland, Maine, course. The weather was overcast and hazy at the beginning of the trial, light to gentle breezes from east southeast, the force of the wind gradually increasing from two to four. At about 9:30 A. M. thick fogs set in and completely obscured the signals on the 19th run, this terminating the standardization trial. The same combinations of turbines were used for corresponding speeds of the standardization trial and the four-hour speed trial, the twelve-hour 24-knot and the twenty-four-hour 16-knot trials.

The principal data of results of the standardization runs are shown in H.P. revolution curve, shown facing this page and on the following table.

FOUR-HOUR FULL-SPEED TRIAL.

(*b*) The four-hour full-speed trial was made on September 6, 1909, beginning at 9:15 A. M. and ending four hours later. The course of the vessel was from Wood Island Light, bearing northwest by west, magnetic, distance four miles, to the Boston Lightship, bearing west northwest, distance eight miles. The weather was clear, with gentle breezes from the northwest to W.N.W., sea very moderate to smooth. The data and results for this trial are as follows:

Draught on trial beginning: Forward, 7 feet 11⅞ inches; aft, 8 feet 4⅝ inches; mean, 7 feet 11½ inches; corresponding displacement, 686 tons.

PERFORMANCE—FOUR-HOURS' OFFICIAL TRIAL, *FLUSSER*.

Steam pressures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Mean steam pressure at boilers, pounds	214.5	
engines, pounds	207.78	
main H.P. turbine (abs.), lbs.	201.89	
L.P. turbine (absolute), lbs.	40.7	40.7
Vacuum in condensers, inches of mercury, mean	28.47	28.29
Auxiliary exhaust (per gauge), pounds	2.61	

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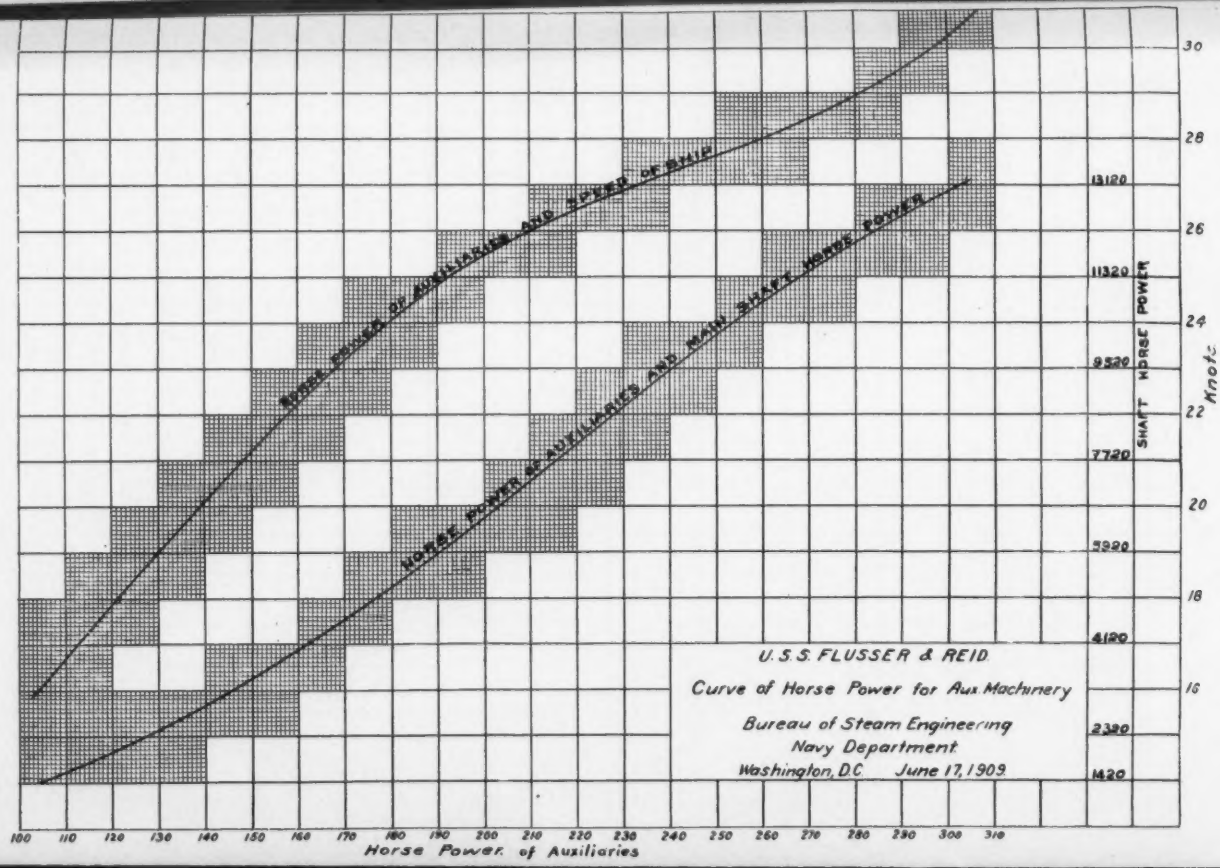
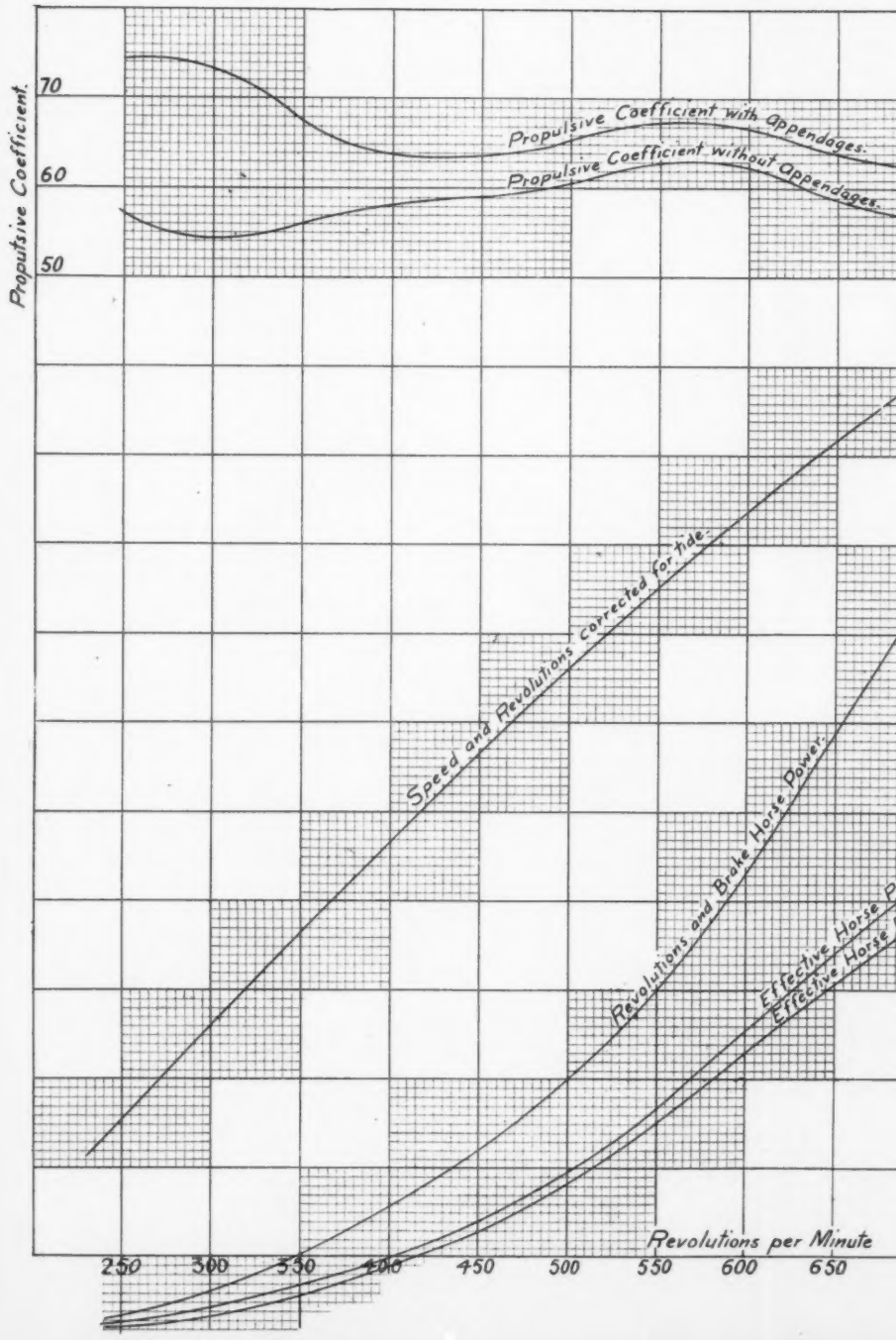

SHOWING POWER OF AUXILIARIES AND POWER OF SHIP
HORSE POWER OF AUXILIARIES AND MAIN SHAFT HORSE POWER
U.S.S. FLUSSER & REID
Curve of Horse Power for Aux. Machinery
Bureau of Steam Engineering
Navy Department
Washington, D.C. June 17, 1909
Horse Power of Auxiliaries
100 110 120 130 140 150 160 170 180 190 200 210 220 230 240 250 260 270 280 290 300 310
SHAFT HORSE POWER
13120
11320
9520
7720
5920
4120
2320
1420
Knots
30
28
26
24
22
20
18
16


Propulsive Coefficient.
70
60
50
Propulsive Coefficient with Appendages.
Propulsive Coefficient without Appendages.
Speed and Revolutions corrected for tide.
Revolutions and Brake Horse Power.
Effective Horse P
Effective Horse
Revolutions per Minute
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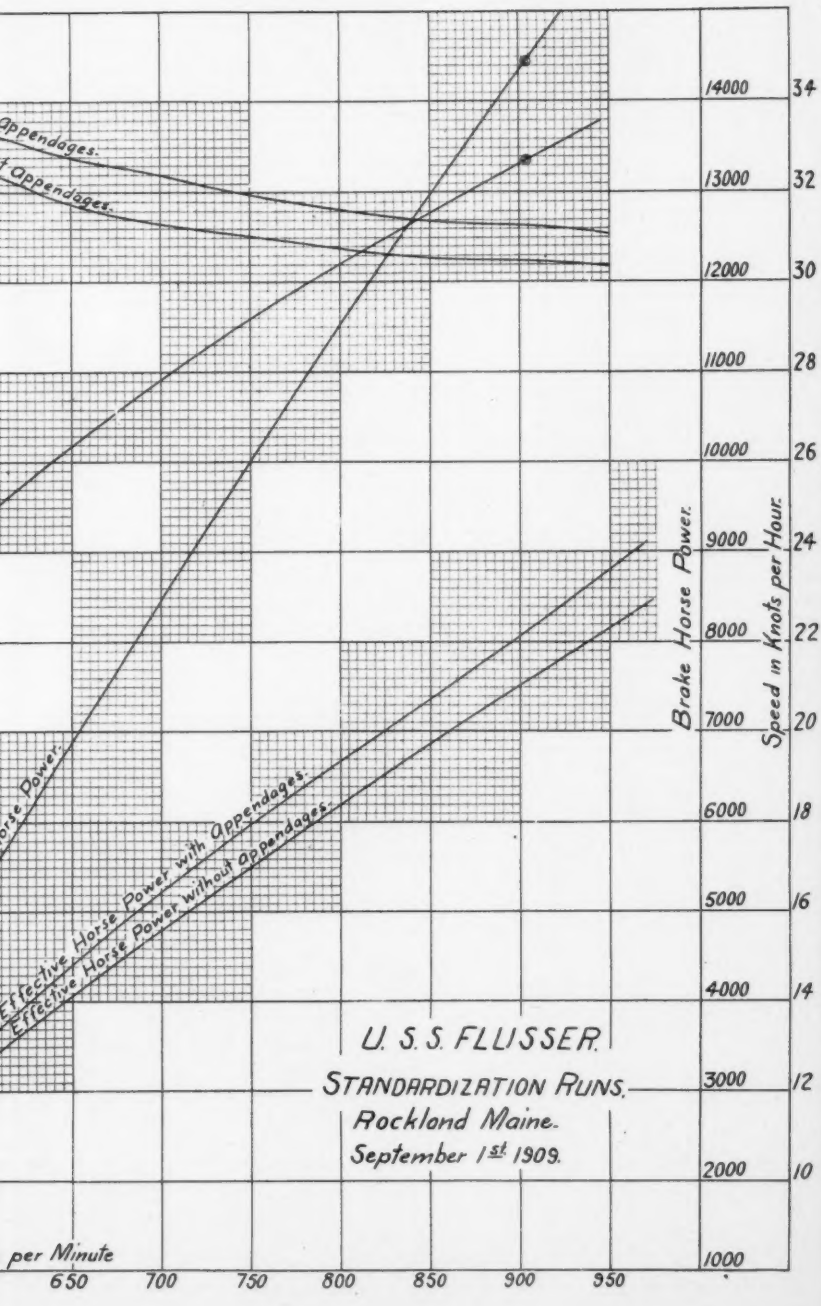
Appendages.
t Appendages.
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Brake Horse Power.
Speed in Knots per Hour.
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16
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12
10
per Minute
650
700
750
800
850
900
950
U. S. S. FLUSSER.
STANDARDIZATION RUNS.
Rockland Maine.
September 1st 1909.

TURBINE DATA (ONE SHEET FOR ENGINE ROOMS) *Standardization* TRIALS

TURBINE DATA (ONE SHEET FOR ENGINE ROOMS) Standardization TRIALS.

(THREE-SHAFT ARRANGEMENT.)

U. S. S. Flusser, Course off Rockland, Maine, on September 1st, 1909.

(FILL IN WITH INDIA INK FOR BLUEPRINTING.)

Trial (6 hours) Sheet 2.

NO. OF RUN.	TIME.			REVOLUTIONS PER MINUTE.				STEAM PRESSURES.										VACUUM IN INCHES OF MERCURY.		TEMPERATURES.												TORSION METER READINGS.			BRAKE HORSEPOWER.			
								ENGINE ROOM (GAGE).			IN STEAM BELT (ABSOLUTE).						ON GLANDS GAGE.	PRESSURE IN OILING SYSTEM (GAGE).		IN ENGINE ROOM.	INJECTION.		DISCHARGE.		AIR-PUMP DISCHARGE.		HEATERS (FEED).				OUTSIDE AIR.	K 1836178	K 1762730	K 1860602				
												CRUISING TURBINES.		L. P. TURBINES.												FEED TO.		FEED FROM. Aft. Forward								
	H.	M.	S.	SHAFT NO. 1.	SHAFT NO. 2.	SHAFT NO. 3.	AVERAGE ALL SHAFTS.	MAIN STEAM.	AUXILIARY STEAM.	AUXILIARY EXHAUST.	MAIN H. P. TURBINE.	H. P.	I. P.	P.	S.			P.	S.		P.	S.	P.	S.	P.	S.	P.	S.	P.	S.		SHAFT NO. 1.	SHAFT NO. 2.	SHAFT NO. 3.	SHAFT NO. 1.	SHAFT NO. 2.	SHAFT NO. 3.	TOTAL.
1	6	39		299.83	299.76	250.99	283.53	250	250	6	12.5	55	25	6.5	6.5	1	6	28.8	28.8	80	54	55	64	62	76	76			230			0.55		.445	303	.55	208	566
2	6	57		277.13	239.71	282.61	266.48	250	245	4	11.5	55	25	5	5	1	5.5	28.8	28.8	79	54	55	62	63	65	72			232			0.34		.375	173	23	197	393
3	7	15		284.95	243.79	281.35	270.03	240	240	5	11.5	55	25	5	5	1	5.5	28.8	28.8	79	54	55	66	63	72	74			230			0.38		.335	199	23	175	397
4	7	30		388.15	318.17	395.91	367.24	245	245	7	23	125	55	8.5	7.5	1	5.5	29.8	29.8	83	55	55	63	63	67	70			230			0.745		.730	530	70	537	1137
5	7	45		387.22	316.85	385.65	363.24	255	255	4	22	125	55	8	7.5	1	5.5	29.8	29.8	81	56	55	64	63	68	70			231			0.745		.795	531	70	570	1171
6	7	57		379.50	317.39	383.43	360.10	257	255	5	20	125	53	7.5	7.5	1	5.5	29.8	29.8	83	56	55	63	63	67	70			230			0.680		.620	453	70	442	965
7	8	10		628.07	550.00	557.54	578.20	245	245	7	85	3.5	2.15	18	20	0	9.	29.5	29.5	83	56	55	67	66	69	72			230	218		2.02	.985	1.41	2329	955	1462	4746
8	8	17		628.99	554.19	562.76	581.98	250	250	3	86	5	2.15	18	20	0	9.5	29.5	29.5	80	56	55	67	66	69	69			219	200		2.027	.985	1.30	2341	962	1361	4664
9	8	27		630.37	554.24	560.35	581.65	250	250	3	87	5	2.15	18	19.5	0	10	29.5	29.5	83	56	55	67	66	70	70			210	210		2.15	1.045	1.32	2488	1021	1376	4885
10	8	37		641.00	642.69	641.55	641.75	240	240	6	135	5	4	25	27	.5	12.5	29	29	80	56	55	71	73	72	72			200	202		1.89	1.83	1.95	2224	2073	2327	6624
11	8	45		641.08	642.97	637.24	640.43	255	255	5.5	135	5	5	25	26	0	12.5	29	29	82	56	55	72	71	71	72			210	186		1.835	1.86	1.95	2162	2108	2312	6582
12	8	54		641.61	636.05	642.91	640.19	250	250	5	130	5	5	26	28	0	11.5	29	29	80	56	55	72	72	72	72			202	190		1.845	1.82	1.97	2173	2040	2356	6569
13	9	00		880.86	879.17	888.44	882.82	245	245	6	200	5	5	37	42	0	15	29	29	82	56	55	80	80	79	78			202	188		2.625	2.90	2.81	4245	4493	4645	13383
14	9	08		900.15	901.45	883.75	895.12	250	250	5	250	5	10	50	50	0	18	28	28	80	56	55	80	77	78	80			192	182		3.04	2.93	2.97	5024	4655	4882	14561
15	9	16		908.40	909.38	916.02	911.27	250	250	5	250	5	10	51	51	0	16	28	28	81	56	55	80	78	78	80			192	210		2.85	3.03	2.935	4753	4856	5002	14611
16	9	22		909.29	910.37	922.19	913.95	250	250	5	252	5	10	51	51	0	11.5	28	28	81	56	55	80	78	79	80			198	202		2.83	2.95	2.86	4724	4734	4906	14364
17	9	30		914.21	912.60	920.64	915.82	250	250	5	252	5	10	51	51	0	15	28	28	82	56	55	80	78	78	80			189	201		2.85	3.11	2.95	4785	5003	5053	14841
18	9	37		728.64	716.99	731.12	725.58	250	250	6	175	5	10	36	36	0	15	28	28	82	56	56	75	74	75	75			188	205		2.41	2.25	2.44	2956	2844	3318	9118
19	9	46																																		
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*TOTAL.																																				
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Temperatures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Injection, degrees	56.0	56.0
Discharge, degrees	76.89	76.89
Air-pump discharge, degrees	77.33	77.33
Feed water, degrees	193.2	194.8
Engine room, working platform, degrees	80.89	
Firerooms, working level, degrees	110.0	

Revolutions, or double strokes, per minute. (*Average of hourly observations.*)

Average revolutions, main engines, per minute, shaft 1,	801.05	
2,	790.46	
3,	812.463	
Mean revolutions, both engines, per minute	801.32	
Pumps, main air	72.22	40.22
feed, d.s., per minute	40.0	
fire and bilge	16.65	
oil	32.61	45.22
Blower engines	820.0	
Speed of ship, in knots per hour	30.41	
Slip of propeller, in per cent. of its own speed, based on mean pitch—Shaft 1, 20.41; shaft 2, 19.34; shaft 3, 21.52; average	20.42	
Air pressure in firerooms, in inches of water, mean	5.0	

Shaft Horsepower.

Shaft No. 1	3,807.75
2	3,695.13
3	4,038.38
total	11,541.26
Total auxiliaries in operation (from curve)	301.0
Collective S.H.P., main engines, plus I.H.P. of auxiliaries	11,842.26

Coal.

Kind and quality used on trial:

Pocahontas, excellent hand-picked, 14,901 B.T.U. per pound.

Pounds per hour, main and auxiliary engines, during trial	22,200
Knots run per ton of coal	3.068

Deduced Data.

S.H.P. per square foot of grate surface	33.23
heating surface	.7134
Pounds of coal per S.H.P. per hour	1.923

Pounds of coal per S.H.P. per square ft. of grate surface, per hour..	63.925
Cooling surface (main condenser), square feet per S.H.P.....	.694

Water.

Pounds of water per hour.....	168,984
H.P. (S.H.P. of shafts plus I.H.P. of aux.).	14.27

TWELVE-HOUR ENDURANCE AND COAL-CONSUMPTION TRIAL AT 24 KNOTS.

The endurance and coal-consumption trial began at 6:30 A. M. and ended at 6:30 P. M., September 4, 1909. The course was from Seguin Island, bearing N.N.E. ½E. magnetic, distance 2.5 miles, and ended off Seguin Island Light, bearing E.N.E. magnetic, distance 5 miles. The weather was fair and hazy, calm at beginning of the trial, and light breezes from the southwest springing up about 7:30 P. M.; from gentle to moderate breezes from S.S.E. about noon; sea smooth up to about noon, then moderate. The courses steered were various, running between Seguin and Thatchers Island. The results for this trial are shown as follows:

Draught on trial, forward, 7 feet 10 inches; aft, 8 feet 4¾ inches; mean, 7 feet 10¼ inches; corresponding displacement, 671 tons.

PERFORMANCE.—TWELVE-HOURS' OFFICIAL TRIAL AT 24 KNOTS (*FLUSSER*).

Steam Pressures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Mean steam pressure at boilers, pounds.....	250.0	
engines, pounds.....	224.77	
main H.P. turbine (abs.), lbs....	89.15	
I.P. cruising turbine (abs.), lbs.	223.46	
L.P. turbine (absolute), lbs.....	42.3	
Vacuum in condensers, inches of mercury, mean.....	29.47	29.66
Auxiliary exhaust (per gauge), pounds.....	4.35	

Temperatures. (*Average of hourly observations.*)

Injection, degrees.....	58.54	58.69
Discharge, degrees.....	69.38	71.08
Air-pump discharge, degrees.....	72.62	72.92
Feed water, degrees.....	203.0	120.62

	Starboard.	*Port.*
Engine room, working platform, degrees	88.92	
Firerooms, working level, degrees	110.0	
Smoke stacks, degrees	540 to 900	

Revolutions, or double strokes, per minute. (*Average of hourly observations.*)

Average revolutions, main engines, per minute, shaft 1,	647.50	
2,	556.50	
3,	576.92	
Mean revolutions, all shafts, per minute	593.64	
Pumps, main air	39.9	37.07
feed, d.s. per minute	35.38	24.7
oil	56.77	12.46
Blower engines	471.5	
Speed of ship, in knots per hour	24.55	
Slip of propeller, in per cent. of its own speed, based on mean pitch. Shaft 1, 20.5; shaft 2, 7.51; shaft 3, 10.78; average	12.93	
Air pressure in firerooms, in inches of water, mean	1.22	

Shaft Horsepower.

Main engines, H.P. cylinder, shaft No. 1	2,678.17
I.P. cylinder, shaft No. 2	962.33
F.L.P. cylinder, shaft No. 3	1,603.17
total three shafts	5,243.67
Total auxiliaries in operation (from curve)	176.0
Collective S.H.P., main engines, plus I.H.P. of auxiliaries	1,280.51

Coal.

Kind and quality used on trial:

Pocahontas, run of mine, 15,064 B.T.U. per pound.

Pounds, per hour, main and auxiliary engines, during trial	8,647.0
Knots run per ton of coal	3.86

Deduced Data.

S.H.P. per square foot of grate surface	15.099
heating surface	0.324
Pounds of coal per S.H.P. per hour	1.65
square foot of grate surface, per hour	24.899
Cooling surface (main condenser), square feet per S.H.P	1.527

Water.

Pounds of water per hour	84,118.0
H.P. (S.H.P. of shafts plus I.H.P. of auxiliaries)	15.52

TWENTY-FOUR-HOUR COAL-CONSUMPTION TRIAL AT 16 KNOTS.

The twenty-four-hour coal-consumption trial at 16 knots was made on September 2 and 3, 1909, beginning at 8 A. M., by chronometer, September 2, and ending at 8 A. M. September 3, 1909. The course was from the entrance to Boothbay Harbor, Maine, and return. The weather was clear and pleasant; light to gentle breezes from west to northwest; smooth sea. Results as follows:

Draught on trial, forward, 7 feet 10⅜ inches; aft, 8 feet 3¾ inches; mean, 7 feet 11⅛ inches; corresponding displacement, 684 tons.

PERFORMANCE.—TWENTY-FOUR HOURS' OFFICIAL TRIAL AT 16 KNOTS.

Steam Pressures. (*Average of hourly observations.*)

	Starboard.	*Port.*
Mean steam pressure at boilers, pounds	216.0	
engines, pounds	215.0	
main H.P. turbine (abs.), lbs.	24.7	
H.P. cruising turbine (abs.), pounds	154.6	
I.P. cruising turbine (abs.), lbs.	66.2	
L.P. turbine (absolute), pounds,	4.46	
Vacuum in condensers, inches of mercury, mean	29.5	29.6
Auxiliary exhaust (per gauge), pounds	5.33	

Temperatures. (*Average of hourly observations.*)

Injection, degrees	57.67	58.20
Discharge, degrees	66.67	67.17
Air-pump discharge, degrees	69.90	69.58
Feed water, degrees	235.0	235.0
Engine room, working platform, degrees	87.9	
Firerooms, working level, degrees	109.56	
Smoke stacks, degrees	520 to 600	

Revolutions, or double strokes, per minute. (*Average of hourly observations.*)

Average revolutions, main engines, per minute, shaft 1,	392.54	
2,	337.37	
3,	386.29	
Mean revolutions, both engines, per minute	372.07	
Pumps, main air	33.31	34.37

	Starboard.	*Port.*
Pumps, circulating	322.0	322.0
feed, d.s., per minute, after pump only	32.35	
fire and bilge	47.33	47.0
oil	36.0	
Speed of ship, in knots per hour	16.26	
Slip of propeller, in per cent. of its own speed, based on mean pitch. Shaft 1, 13.15; shaft 2, 1.05; shaft 3, 11.75; average	8.0	
Air pressure in firerooms, in inches of water, mean	0.512	

Shaft Horsepower.

Shaft No. 1	521.17
2	159.17
3	494.17
Total three shafts	1,174.51
auxiliaries in operation (from curve)	106.0
Collective S.H.P. main engines plus I.H.P. of auxiliaries	1,280.51

Coal.

Kind and quality used on trial:
Pocahontas, run of mine, 14,856 B.T.U. per pound.

Pounds per hour, main and auxiliary engines, during trial	2,678.0
Knots run per ton of coal	13.61

Deduced Data.

S.H.P. per square foot of grate surface	6.764
heating surface	0.145
Pounds of coal per S.H.P. per hour	2.28
Pounds of coal per square foot of grate surface, per hour	15.42
Cooling surface (main condenser), square feet per S.H.P.	6.816

Water.

Pounds of water per hour	32,244.0
H.P. (S.H.P. of shafts plus I.H.P. of auxiliaries)	25.168

AUXILIARY MACHINERY ON TRIALS.

The auxiliary machinery in operation during the full-speed four-hour trial was as follows: Both main air pumps, both main feed pumps, all fireroom forced-draft blowers, both lubricating oil-circulating pumps, one fire and bilge pump, intermittently, and the electric-light plant system, consisting

of one Curtis turbine with a 5-kw. generator, making 5,000 revolutions per minute, 120 volts and 40 ampères.

During the 24-hour 16-knot trial the auxiliaries were run as follows:

(*a*) Auxiliaries in connection with the operation of the main engines, including feed pumps.

(*b*) Dynamo as required to efficiently light the ship.

(*c*) The evaporator and distilling plant necessary to supply "make-up" feed water and water for all other purposes incident to conditions.

(*d*) Flushing system was in continuous operation.

(*e*) Steering engine was used as required for steering.

During the 24-hour 12-knot trial the following auxiliaries were in operation:

(*a*) Auxiliaries in connection with the operation of the main engine, including feed pump.

(*b*) Dynamo as necessary to efficiently light the ship.

(*c*) The evaporating and distilling plant in continuous operation at normal rate of capacity.

(*d*) Flushing system in continuous operation.

(*e*) Forced-draft fans as required (one in each fireroom).

(*f*) Steering engine as required for steering.

COAL CONSUMPTION.

16.20 knots, 24-hour run, 820 bags, 29.28 tons = 29.28 tons a day.
24.55 knots, 12-hour run, 1,292 bags, 46.14 tons = 92.28 tons a day.
30.41 knots, 4-hour run, 1,110 bags, 39.64 tons = 237.84 tons a day.

After the standardization trial and with the vessel steaming at approximately 28 knots, the helm being exactly amidships, the order was given to put the helm hard starboard. The wheel was put over in 9.5 seconds; the vessel swung around under hard-starboard helm until she completed a circle in 2 minutes and 21.8 seconds. The turning of the circle was estimated to be about 500 yards and the angle of heel about three degrees.

MANEUVERING TEST.

With the helm amidships and the vessel making approximately 28 knots, the engine telegraphs were thrown into full speed astern; the engines were promptly reversed, and the ship was dead in the water at 27.4 seconds. The indicators and telegraphs were then thrown to full speed ahead; the engines answered the signals instantly.

The steering gear was then tested at reduced speed while the vessel was going astern; the helm being amidships, the mechanical telegraphs on the bridge were brought half speed astern. When the engines were turning over astern at the rate of 420 revolutions per minute the helm was brought successively to 15 starboard, 25 port, 30 starboard, 35 port, 35 starboard, and allowed to remain in each position as long as practicable, considering the thick fog and proximity to land. All these tests resulted satisfactorily.

POST-TRIAL EXAMINATION.

On the examination of the machinery after the full-speed four-hour trial, for which all the turbines were opened, the rotors and casing blading examined, all parts were found in good condition. The boilers, which were likewise examined on their water sides, were found in excellent condition; no grease or sediment of any kind was present. The condensers were found to be tight. Two fireroom auxiliary feed-pump rods were found to be scored, and were renewed. The main H.P. thrust bearing was somewhat scored and the brass collars smeared. In general, the condition of the machinery after the trial was found to be very good. Two bolts securing the boiler casings to the lower drums were found in a slightly leaky condition and were renewed by the contractors. Some of these were afterward renewed for the contractors at the Navy Yard, Boston, after delivery of the vessel.

OFFICIAL TRIALS OF "REID."

The standardization trial was begun at 6:08 A. M. on October 6, 1909, and finished at 9:04 A. M. the same day.

The weather was fair, with light airs from W.N.W.; smooth sea.

The results of these runs are shown in speed and revolution curves accompanying, and the principal data is given in the attached table.

FOUR-HOUR FULL-SPEED TRIAL.

The *Reid* left for this trial at 12:42 P. M., October 9, 1909, and the trial began at 1:40 P. M. off Sequin Island Light, and was finished four hours later off Portland Lightship. The weather was partly cloudy; light breezes from S.S.W. to S.W. Courses various; smooth sea.

The results of trial are as follows:

Draught on trial: Forward, 8 feet $\frac{1}{16}$ inch; aft, 8 feet $6\frac{9}{16}$ inches; mean, 8 feet; corresponding displacement, 690 tons.

PERFORMANCE—FOUR-HOURS' OFFICIAL TRIAL (*REID*).

Steam Pressures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Mean steam pressure at boilers, pounds	232.5	
engines, pounds	218.0	
H.P. turbine (absolute), lbs	216.2	
L.P. turbine (absolute), pounds,	47.10	48.28
Vacuum in condensers, inches of mercury, mean	28.67	28.54
Pressure in auxiliary exhaust, gauge, pounds	5.8	

Temperatures. (*Average of one-half hourly observations.*)

Injection, degrees	58.7	57.9
Discharge, degrees	77.9	76.7
Air-pump discharge	79.3	79.3
Feed water, degrees	192.7	197.9
Engine room, working platform, degrees	101.8	
Firerooms, working level, degrees	91.8	
Smoke stacks, degrees	685-985	

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

Average revolutions, main engines, per min., shaft 1	852.56
2	826.50
3	848.83
Mean revolutions, all shafts, per minute	842.63

TURBINE DATA (ONE SHEET FOR ENGINE ROOMS) STANDARDIZATION TRIALS.

(THREE-SHAFT ARRANGEMENT.)

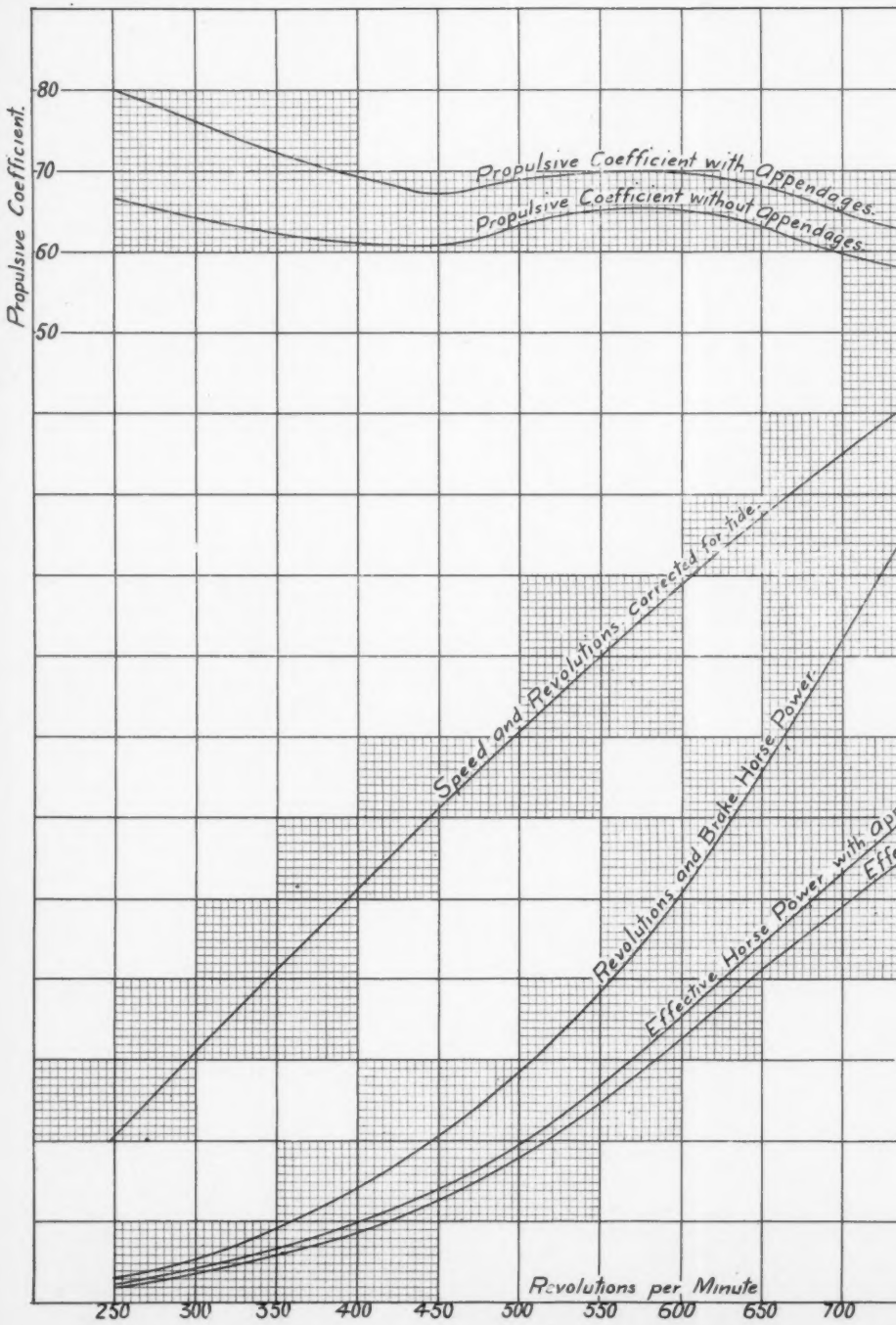
U. S. S. REID, Course off Rockland, Maine, on October 6, 1905

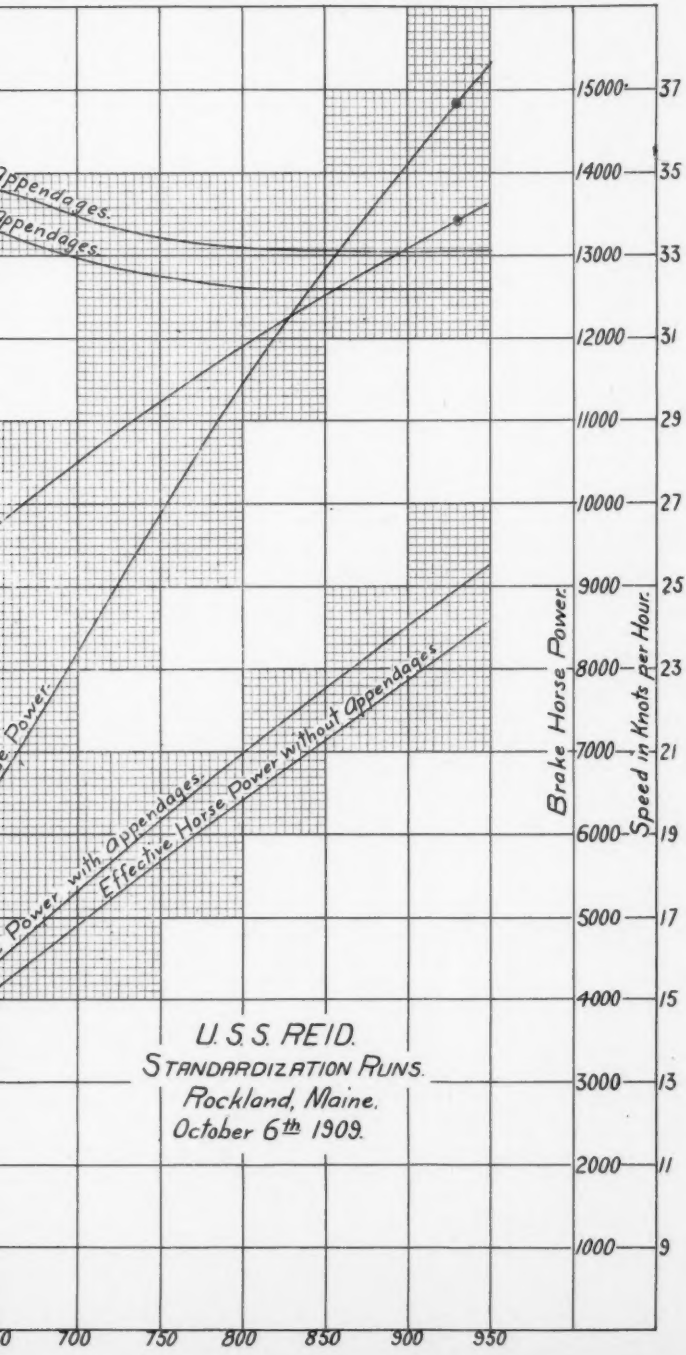
(FILL IN WITH INDIA INK FOR BLUEPRINTING.)

Trial (2) Data. Sheet 2.

NO. OF RUN.	TIME. H.	M.	S.	REVOLUTIONS PER MINUTE. SHAFT NO. 1.	SHAFT NO. 2.	SHAFT NO. 3.	AVERAGE ALL SHAFTS.	STEAM PRESSURES. ENGINE ROOM (GAGE). MAIN STEAM.	AUXILIARY STEAM.	AUXILIARY EXHAUST.	IN STEAM BELT (ABSOLUTE). - MAIN H. P. TURBINE.	CRUISING TURBINES. H.P.	I.P.	L. P. TURBINES. P.	S.	ON GLANDS GAGE.	PRESSURE IN OILING SYSTEM (GAGE).	VACUUM IN INCHES OF MERCURY. P.	S.	TEMPERATURES. IN ENGINE ROOM.	INJECTION. P.	S.	DISCHARGE. P.	S.	AIR-PUMP DISCHARGE. P.	S.	HEATERS (FEED). FEED TO.		FEED FROM.		OUTSIDE AIR.	TORSION METER READINGS. K=1.24805 SHAFT NO. 1.	K=1.21342 SHAFT NO. 2.	K=1.27440 SHAFT NO. 3.	BRAKE HORSEPOWER. SHAFT NO. 1.	SHAFT NO. 2.	SHAFT NO. 3.	TOTAL.
1	6	08	50	279.27	256.85	279.61	271.91	225	230	11	11	55	26	5	4	0	5	29.6	29.3	105	54	55	62	63	69	70	70		214	240		.21	.28	.26	108	127	136	371
2	6	20	30	292.06	264.58	279.88	278.88	230	230	10	11	55	26	5	4	0	5	29.6	29.4	104	54	55	62	63	74	71	74		218	242		.30	.29	.25	161	135	131	427
3	6	32	00	270.70	237.95	274.03	260.89	234	232	3	10	55	26	5	3.5	.2	5.1	29.6	29.3	103	56	55	68	66	76	75	76		218	240		.17	.19	.27	84	79	139	302
4	6	43	45	389.60	345.12	378.11	370.94	255	254	9	11.5	130	58	7.5	6.5	0	6	29.6	29.3	100	55	55	66	65	65	70	75		227	250		.60	.45	.46	430	274	326	1030
5	6	52	28	376.28	340.51	376.24	365.01	242	242	7	11.5	130	57	7	6.5	0	5.3	29.6	29.3	99	56	55	65	65	73	70	73		228	250		.62	.47	.52	432	282	367	1081
6	7	02	12	387.31	346.81	378.03	370.72	242	242	12	11	130	57	7.5	6.5	0	5.1	29.6	29.3	100	56	55	65	65	72	70	72		234	255		.55	.48	.49	392	294	347	1033
7	7	13	32	636.97	558.91	564.97	586.95	242	245	6	60	2	215	18.5	17	0	8	29.2	29.1	99	55	55	66	68	71	70	71		234	250		1.83	1.36	1.07	2142	1341	1133	4616
8	7	21	35	636.14	560.06	555.22	583.81	254	253	4.5	80	2.5	215	17.5	17	0	8	29.5	29.1	98	55	55	66	67	71	68	71		234	260		1.94	1.32	1.02	2270	1304	1061	4635
9	7	29	16	640.82	555.70	574.13	590.22	251	252	11	80	2.5	215	17.5	18.5	0	8	29.5	29	98	55	55	66	68	70	70	70		214	260		1.94	1.33	1.17	2287	1304	1259	4850
10	7	36	38	836.71	805.43	820.18	820.77	251	250	7	205	2.5	2	45	45	0	8	28.5	27.8	99	55	55	73	76	77	77	70		230	240		2.39	3.03	2.44	3680	4305	3749	11734
11	7	43	17	796.66	776.32	798.52	790.50	250	250	9	191	.5	2	42	42	0	8	28.4	27.8	100	56	55	73	76	77	77	76		205	152		2.43	2.98	2.42	3562	4080	3622	11264
12	7	49	54	812.24	782.35	794.75	796.45	253	255	10	195	2.5	2.5	43.8	42	0	8.5	28.4	27.9	101	56	55	73	76	77	77	77		200	210		2.45	3.00	2.44	3664	4140	3634	11438
⊕13	7	57	10	901.68	880.84	902.50	895.01	230	232	10	235	2.5	2.5	53	52	0	8	28.1	27.3	101	56	55	72	78	78	79	77		212	174		2.77	3.25	2.66	4596	5049	4500	14145
14	8	04	21	949.90	926.05	934.66	936.87	252	255	12	256	2.5	3.5	57	57	0	9	28	27.4	102	56	55	72	78	79	79	78		190	200		2.82	3.26	2.73	4924	5325	4783	15032
15	8	11	01	931.35	913.59	932.83	925.92	250	255	9	248	2.5	3.5	55	55	0	10	28	27.4	103	56	55	72	78	78	79	78		210	200		2.83	3.12	2.72	4850	5028	4756	14634
16	8	17	18	944.71	920.72	930.48	931.97	250	255	10	257	2.5	3.5	57	57	0	10	28	27.3	103	56	55	73	78	79	79	79		210	200		2.84	3.23	2.76	4936	5244	4814	14994
⊕17	8	23	50	868.51	839.17	868.34	858.67	252	260	12	256	2.5	3.5	57	56	0	10	28	27.3	102	56	55	73	79	78	78	78		211	190		2.83	3.11	2.76	4603	4603	4492	13698
18	8	31	02	753.81	730.32	739.91	741.35	255	257	7	170	2.5	4	41.5	38	0	9.5	28.7	28.2	101	56	55	71	74	74	74	74		215	185		2.24	2.60	2.32	2695	3350	3217	9262
19	8	38	45	725.59	714.75	726.11	722.15	240	240	4	161	2.5	4	36	36	0	9	28.6	28.2	101	56	55	72	74	74	74	74		210	185		2.16	2.57	2.11	2883	3240	2872	8995
20	8	46	13	725.98	713.65	711.65	717.09	230	235	3	161	2.5	4	36	35	0	9	28.6	28.2	100	56	55	71	77	74	74	74		209	185		2.18	2.50	2.17	2912	3146	2895	8953
⊕21	8	55	00	674.40	663.08	674.89	670.79	230	235	6	115	2.5	2.5	26	25	0	11	28.6	28.2	101	56	55	71	73	73	73	73		219	185		2.27	2.38	2.09	2817	2784	2644	8245
⊕22	9	02	00	722.36	709.94	708.63	713.64	246	248	7	160	2.5	4.5	35	35	0	11.5	28.8	28.2	100	56	55	71	74	74	74	74		210	245		2.19	2.48	1.96	2911	3106	2603	8620
23																																						
24																																						
25																																						
TOTAL																																						

V
1
E
0
XU


Propulsive Coefficient.
80
70
60
50
Propulsive Coefficient with Appendages.
Propulsive Coefficient without Appendages.
Speed and Revolutions, Corrected for tide.
Revolutions and Brake Horse Power.
Effective Horse Power with Ap
Eff
Revolutions per Minute
250
300
350
400
450
500
550
600
650
700


Appendages.
Appendages.
Power.
Power with Appendages.
Effective Horse Power without Appendages.
Brake Horse Power.
Speed in Knots per Hour.
15000
14000
13000
12000
11000
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
37
35
33
31
29
27
25
23
21
19
17
15
13
11
9
700
750
800
850
900
950
U. S. S. REID.
STANDARDIZATION RUNS.
Rockland, Maine.
October 6th 1909.

	Starboard.	Port.
Pumps, main air	76.0	57.6
circulating, auxiliary	611.0	
feed, d.s., per minute	f. 48.4	a. 37.3
oil	44.8	
Blower engines	642.0	
Speed of ship, in knots per hour	31.82	
Slip of propeller, in per cent. of its own speed, based on mean pitch—Shaft 1, 21.7; shaft 2, 19.2; shaft 3, 21.4; mean	20.8	
Air pressure in firerooms, in inches of water, mean	5.19	

Shaft Horsepower.

Main engines, shaft No. 1	4,224.0
2	4,268.0
3	3,929.0
total, all shafts	12,421.0
Total, all auxiliaries in operation (from curve)	313.0
Collective S.H.P., main engines, plus I.H.P. of auxiliaries,	12,734.0

Coal.

Kind and quality used on trial:

Pocahontas, excellent, hand-picked, 15,046 B.T.U. per pound.

Pounds per hour, main and auxiliary engines, during trial	19,880.0
Knots run per ton of coal	3.6

Deduced Data.

S.H.P. per square foot of grate surface	35.77
main engines, per square foot of heating surface	0.768
Pounds of coal per S.H.P. per hour	1.6
square foot of grate surface, per hour	57.24
Cooling surface (main condenser), square foot per S.H.P.	0.645

Water.

Pounds of water per hour	183,371.0
H.P. (S.H.P. plus I.H.P. of auxiliaries)	14.40

TWELVE-HOURS' TRIAL AT 24 KNOTS.

The *Reid* left for this trial at 9:10 A. M., October 8, 1909, and the trial began at 10:09 A. M., off Seguin Island Light, and ended twelve hours later off Seguin Island Light. The weather was fair and hazy, generally calm, with smooth sea.

The results of this trial are as follows:

Draught on trial: Forward, 8 feet $3\frac{1}{2}$ inches; aft, 8 feet $7\frac{3}{8}$ inches; mean, 8 feet $1\frac{3}{4}$ inches; corresponding displacement, 713 tons.

PERFORMANCE—TWELVE-HOURS' OFFICIAL TRIAL AT 24 KNOTS (*REID*).

Steam Pressures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Mean steam pressure at boilers, pounds	248.0	
engines, pounds	247.0	
H.P. turbine (absolute), pounds,	82.8	
L.P. turbine (absolute), pounds,	19.08	20.25
H.P. cr. turbine (abs.), pounds..	19.0	
I.P. cr. turbine (abs.), pounds...	225.0	
Vacuum in condensers, inches of mercury, mean	29.65	29.26
Auxiliary exhaust, per gauge, pounds	9.1	

Temperatures. (*Average of hourly observations.*)

Injection, degrees	56.0	56.0
Discharge, degrees	69.0	68.0
Air-pump discharge, degrees	71.0	72.0
Engine room, working platform, degrees	98.0	
Firerooms, working level, degrees	104.5	

Revolutions, or double strokes, per minute. (*Average of hourly observations.*)

Average revolutions, main engines, per minute, shaft 1	659.08	
2	560.77	
3	582.32	
Mean revolutions, all shafts, per minute	600.72	
Pumps, main air	53.5	56.2
circulating, auxiliary	383.0	
feed, d.s., per minute	f. 35.1	a. 28.6
oil	in 35.5	out 41.4
Blower engines	404.0	
Speed of ship, in knots per hour	24.72	
Slip of propeller, in per cent. of its own speed, based on mean pitch—Shaft 1, 21.69; shaft 2, 7.57; shaft 3, 11.0; mean	13.42	
Air pressure in firerooms, in inches of water, mean	1.46	

Shaft Horsepower.

Shaft No. 1	2,708.0
2	1,359.0
3	1,346.0
Total, all shafts	5,413.0
all auxiliaries in operation (from curve)	189.0
Collective S.H.P., main engines, plus, I.H.P. of auxiliaries	5,602.0

Coal.

Kind and quality used on trial:
Pocahontas, run of mine, 15,073 B.T.U. per pound.
Pounds, per hour, main and auxiliary engines, during trial..... 9,913
Knots run, per ton of coal..... 5.59

Deduced Data.

S.H.P. per square foot of grate surface..... 15.59
main engines, per square foot of heating surface..... 0.335
Pounds of coal per S.H.P. per hour..... 1.83
square foot of grate surface, per hour..... 28.54
Cooling surface (main condenser), square feet per S.H.P..... 1.479

Water.

Pounds of water per hour..... 82,520.0
H.P. (S.H.P. plus I.H.P. of auxiliaries)..... 14.73

TWENTY-FOUR-HOUR TRIAL AT 16 KNOTS (*REID*).

The *Reid* left for this trial at 6:08 P. M., October 6, 1909, and the trial began at 6:29 P. M., off Owls Head, Rockland, and ended twenty-four hours later off Seguin Island. The courses steered were various, running between Monehegan Island and the Whistling Buoy off Cape Ann. The weather was fair, with a light haze. Calms and light breezes from N.W. to S., smooth sea.

The results of this trial were as follows:

Draught at trial: Forward, 7 feet 9¾ inches; aft, 8 feet 8¼ inches; mean, 8 feet ⅞ inch; corresponding displacement, 703 tons.

PERFORMANCE—TWENTY-FOUR-HOURS' OFFICIAL TRIAL AT 16 KNOTS (*REID*).

Steam Pressures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Mean steam pressure at boilers, pounds.....	243.0	
engines, pounds.....	235.0	
H.P. turbine (absolute), pounds,	22.5	
L.P. turbine (absolute), pounds,	7.72	8.85
H.P. cr. turbine (abs.), pounds..	143.0	
I.P. cr. turbine (abs.), pounds...	62.0	
Vacuum in condensers, inches of mercury, mean.....	29.3	29.16
Auxiliary exhaust, per gauge, pounds.....	10.3	

Temperatures. (*Average of hourly observations.*)

	Starboard.	Port.
Injection, degrees	55.0	55.0
Discharge, degrees	68.3	68.2
Air-pump discharge, degrees	71.8	73.0
Feed water, degrees	92.8	
Engine room, working platform, degrees	99.0	
Firerooms, working level, degrees	114.0	

Revolutions, or double strokes, per minute. (*Average of hourly observations.*)

Average revolutions, main engines, per minute, shaft 1	392.36	
2	347.60	
3	398.88	
Mean revolutions, all shafts, per minute	379.35	
Pumps, main air	48.3	48.9
circulating, auxiliary	416.0	134.0
feed, one, d.s., per minute	23.4	
oil	46.7	
Speed of ship, in knots per hour	16.40	
Slip of propeller, in per cent. of its own speed, based on mean pitch—Shaft 1, 12.37; shaft 2, 1.07; shaft 3, 13.81; mean	9.08	
Air pressure in firerooms, in inches of water, mean	.782	

Shaft Horsepower.

Shaft No. 1	493.0
2	278.0
3	427.0
Total all shafts	1,198.0
auxiliaries in operation (from curve)	108.0
Collective S.H.P. main engines plus I.H.P. of auxiliaries	1,306.0

Coal.

Kind and quality used on trial:

Pocahontas, run of mine, 15,228 B.T.U. per pound.

Pounds per hour, main and auxiliary engines, during trial	3,410.0
Knots run per ton of coal	10.77

Deduced Data.

S.H.P. (total) per square foot of grate surface	6.9
main engines, per square foot of heating surface	0.1481
Pounds of coal per S.H.P. per hour	2.846
square foot of grate surface, per hour	19.64
Cooling surface (main condenser), square feet per S.H.P	6.682

Water.

Pounds of water per hour	30,342.0
H.P. (S.H.P. plus I.H.P. of auxiliaries)	23.231

The same auxiliaries as in the case of the *Flusser* (page 36) were in operation during the various trials.

ATTACK OF WARSHIPS AND TABLES OF FIRE EFFECT.*

By Captain John W. Gulick, Coast Artillery Corps.

PART I.

1. However interesting and instructive a study of results of tests of armor and projectiles on the proving ground may be, the practical coast artilleryman is chiefly concerned with the behavior of modern armor and projectiles under service or battle conditions.

What will the probable battle angles of impact be? Can perforation of main belt and turret armor be expected at modern battle ranges and under service conditions? What is the best type of projectile and what should be the character of its bursting charge? All of these questions are of vital interest to the practical officer, and in the following discussion an attempt will be made to approach these questions in a very practical manner.

2. *Battle Angles of Impact.*—There are as many diverse views as to what is the average battle angle of impact as there are authorities. It is now generally agreed, however, that less than ten per cent. of the shots fired in action at a moving ship in a moderate sea will hit the target normally, and, considering impact with reference to the horizontal and vertical planes, it will easily be seen that at any range normal impact is out of the question. For high-sited batteries impact will always be oblique, and for those batteries so located as to give a "head on" or raking fire on approaching ships, the target is presented in such a manner as to be unfavorable for securing anything approaching normal impact. In this connection some remarks

* Extracts from paper published in Journal of the United States Artillery.

by Mr. Jane, which appeared in "Fighting Ships," 1907, are of interest:

"The original turret was made circular so as to deflect shot by the angle presented. Subsequently very inclined structures came in, the object being to deflect projectiles. The barbettes of ships like the *Sicilia*, and the slope of armor decks represent this principle fully applied. Then at some time a belief that projectiles struck point downwards appears to have affected construction, for wall-sided turrets reappeared. The fact that a ship is probably rolling and that therefore, except in extreme cases, armor, however placed, only by chance can be at right angles to the projectile no doubt also entered. In the matter of turrets France and Russia generally tend to adhere to the original circular turret, made oval for reasons of reduction of size. The United States and Germany mostly go in for the oval turret with inclined and rounded sides. British turrets are chiefly remarkable for varying with almost every class of ship. The old oval, sloping, round-sided turret was, it is said, driven out of existence by the engineers' strike—the necessary men were not available to construct these particular hoods. This evolution by necessity is believed to be the real origin of the flat-sided turret, first found in the *London*, the present special advantage of which is that K.C. plates can be used. These turrets are very multi-sided; the *Swiftsure* and *Triumph* appeared with simpler six-sided forms of these. The *Black Prince* class embody the same type. In the *Dreadnought* type another variation appears, the back of the turret being raised considerably. A projectile on striking is invariably parallel to the muzzle of the gun. Therefore on a quite stationary target at short range it strikes at right angles, and at long range it strikes point upward. *For instance, supposing the range to require a 20-degree elevation to the gun, the projectile will strike vertical armor at 20 degrees from normal. A turret with sides normally inclined at 20 degrees will be struck at 40 degrees, and so on.*

"Were the ship rolling 10 degrees, if towards the gun, the normally vertical armor will be struck at 10 degrees, while if

rolling away at the moment of impact, the impact will be 30 degrees from the normal, and all of this is without regard to the lateral angle that the ship may also present. The chances of an absolutely normal hit are therefore extremely small even on a vertical belt, upon an inclined turret they are non-existent unless the ship is rolling abnormally."

The conditions outlined above in italics, were fully discussed by Captain Tressider, in a paper which was read before the Institution of Naval Architects, British, and reprinted in the "Journal U. S. Artillery," Whole No. 91. The oblique impact generally recognized has been that of the inclination of the trajectory to the normal to the plate, the blow of the projectile being fairly delivered through its point. In the oblique impact referred to above the axis of the projectile is inclined to the trajectory, while the latter may or may not be inclined to the

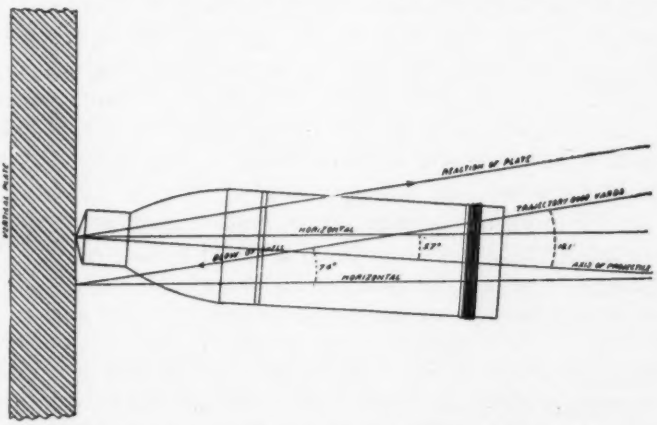


Fig. 1.—TRESSIDER'S THEORY OF GYROSTATIC ACTION, ETC.

normal to the plate. It is assumed that the gyrostatic action due to axial rotation, though reduced by air friction, remains powerful to the last and retains the projectile's axis pointing upwards when impact takes place. Thus, in the case of a 12-inch A.P. shell, capped, loaded and fused, fired at vertical armor at 8,000 yards, it leaves the muzzle with its axis point-

ing upwards at an angle of 5.7 degrees, and when impact takes place the trajectory is pointing downwards at an angle of 7.4 degrees. The figure illustrates this condition, and it will easily be seen that the angle 5.7 plus 7.4 or 13.1 degrees between the axis and the trajectory is much more unfavorable for the projectile than a larger angle would be between the trajectory and normal to the plate. The "couple" formed by the momentum of the shell acting along its trajectory and through its center of gravity, and the reactance of the plate acting in an opposite parallel direction through the point of impact, must have a powerful cross-breaking effect on the shell.*

If the roll of the ship brings its side armor normal to the trajectory at the instant of impact, it increases the obliquity between the axis of the projectile and the normal from 5.7 to 13.1 degrees, and if the roll of the ship is in the other direction and brings the shell's axis into the normal it increases the obliquity of the trajectory to 13.1 degrees. On the whole it would appear that normal impact at a range is not only unlikely, but may be actually impossible. Also that the defensive power of armor under battle conditions will be greater than is indicated by tests with the same velocities, at short range on the proving ground.

3. *Perforation of Armor Under Battle Conditions.*—Modern battle ranges will vary from 6,000 to 10,000 yards, and taking into account the probable battle angle of impact which we place at 60 degrees, the most powerful direct-fire guns will be unable to accomplish anything serious by penetration of the main belt armor except at the lightly armored ends, when the enemy exposes his broadside. With the turrets and barbettes, which to some extent face in all directions and are protected by armor equivalent to 12 inches of K.C., no gun in our system can be relied on to perforate these structures at 6,000 yards.

The Russo-Japanese war was the first in which modern battleships and modern guns met in hostile opposition. From

*The Bureau of Ordnance, U. S. Navy Department, at this date are preparing to make a practical test of this theory. Firings are to be conducted at an armored structure at long range, in which the inclination of the axis of the projectile upon arrival at the point of impact will be definitely determined.

a careful study of the naval actions of this war, based upon the best data obtainable, it would appear that modern battleships provided with excellent armor protection may be put completely out of action by gun fire alone in a very short period of time, and that this may be accomplished without perforation of main armor. The following statements are of interest:

(1) There is no record of a single case in which the main belt armor was perforated, either at Tsushima or in the battle of August 10th, 1904.

(2) As in the case of belt armor, we have no reliable information that any heavy turret and casemate armor was perforated. The following table which originally appeared in the "Artilleristiche Monatshefte," contains most of the available information in respect to serious hits on such armor.

TABLE.

No.	Ship.	Type of target.	Thickness and type of armor.	Projectile.		Effect on the armor.
				Type.	Caliber.	
			mm.			
1	*Tsarevitch.*	30.5 cm. turret.	254 Krupp steel.	Shell.	30.5	Did not perforate. Surface flaked out.
2	*Tsarevitch.*	30.5 cm. turret.	254 Krupp steel.	High exp. shell.	30.5	Burst without effect.
3	*Tsarevitch.*	Conning tower.	254 Krupp steel.	Shell with cap.	30.5	Did not perforate. The shot deflected entered through an opening and burst there; 56 men killed, apparatus disabled. The head of projectile passed outside through another opening, and later found in the hammock nettings.
4	*Tsarevitch.*	Cover of 30.5 cm turret near the sighting cap.	254 Krupp steel.	High exp. shell.	Unknown.	Did not perforate. Cover slightly bent. Exploded fragments killed one man and seriously wounded another. The man in the sighting hood not hurt.
5	*Orel.*	30.5 cm. turret.	254 Krupp steel.	Unknown.	Unknown.	Did not perforate. Surface splintered 3 mm. thick. Thick protecting shield over the piece bent, lowering the range to 731 m. Nothing broken.
6	*Orel.*	30.5 cm. turret.	254 Krupp steel.	Shell.	30.5	Did not perforate. Surface slightly splintered.
7	*Orel.*	15 cm. turret.	152 Krupp steel.	Shell.	Unknown.	Did not perforate. Two places hit. One plate 150 mm. thick bent inward nearly 90 degrees. Surface pressed backward, but not splintered.
8	*Orel.*	Rear casemate.	76 cast steel.	Two shells	15 and 20.3	Did not perforate.

The main point to be remembered, is that armor of any kind may be expected to give a much better account of itself under battle conditions than on the proving ground. This point is well illustrated in some experiments recently conducted by the British Admiralty, using the old battleship *Hero* as the target. Built in 1888, the *Hero* carried fairly modern armor (compound), comprising 12-inch plates on her water line, 12 inches on the turret containing the main armament of two 12-inch guns, and 11.5 inches on the bulkhead. The armor belt tapered from a maximum of 12 inches at the water line to 8.5 inches, and extended to within six fathoms of the stern. The *Hero* was moored in an east and west direction, so that she presented her full broadside to the attacking ships. The battleship squadron of the Channel fleet, consisting of the *King Edward VII*, flagship, the *Hibernia*, the *Illustrious*, the *Dominion*, the *Ocean*, the *Jupiter* and the *Vengeance*, steamed past the *Hero* very slowly, closing up from 8,000 to 6,000 yards. The firing ships were the *Hibernia* and *Dominion*, both recent additions to the British Navy. Thier displacement is 16,500 tons, and the armament consists of four 12-inch guns, four 9.2-inch guns and ten 6-inch guns, all of which were used in the test. One witness in describing the test, says: "Very soon the *Hero* was in the middle of a dense cloud of smoke, through which, now and again, flashed great tongues of fire, as shell after shell struck the old ship. It was soon over, however, and save for the slight list to port, there was nothing to show at a distance the ordeal through which she had passed. The funnel and mast both stood, and very little smoke issued from the interior. The upper deck had been blown up by the explosion of a heavy shell underneath it, and the after conning tower had been penetrated. Altogether twenty-eight hits had been scored out of one hundred and thirty rounds." It has been stated that the old compound armor with its F.M. of about 1.25, gave an excellent account of itself and was not perforated, though receiving the majority of the hits.

Until recently the attack of deck armor by direct-fire guns received a good deal of attention in our service, and our

literature was full of such terms as "deck-piercing zones," etc. It is now, however, generally conceded that the attack of armored decks with direct-fire guns, with an idea of perforating the deck is impracticable. The results of some experiments recently conducted by the U. S. Army Ordnance Board at Sandy Hook would appear to indicate that perforation of armored decks at battle ranges is not to be expected. For attack of such decks, with a view of reaching the vitals of a modern armored ship, the 12-inch B.L. mortar is the best weapon in our system. The angles of impact will be favorable for perforation of armored decks of 4 inches of unhardened steel or its equivalent at all ranges beyond 4,500 yards, and as is well known the accuracy of the mortar compared favorably with direct-fire guns of large caliber at extreme ranges.

4. *Best Type of Projectile.*—The lessons that may be deduced from a careful study of the naval actions of the Russo-Japanese war have had a decided effect in shaping policies of the different powers, with special reference to the character or type of projectile. These lessons, however, have been interpreted differently, and, while applying particularly to naval ordnance, they are of interest to the coast artilleryman, as the main problem is the same. Some authorities are in favor of a common steel shell, with little or no perforating power, carrying a bursting charge of high explosive equal to 10 or 12 per cent. of its weight, and equipped with a non-delay-action fuse. Others are in favor of a semi-armor piercing shell, such as our A.P. shell or the semi-rupture shell of the French and Germans, carrying a high explosive bursting charge equal to 6 per cent. of its weight, and equipped with a delay-action fuse, and capable of perforating on the proving ground armor one-half caliber thick.

In both of the policies outlined above it is recognized that the perforation of armor is not the primary consideration, but that the main object is to inflict the maximum number of damaging hits in the shortest time. The modern battleship presents a large target, of which only a portion can carry thick armor. The experience of the battle of Tsushima plainly

shows, if it shows nothing else, that the perforation of thick belt and turret armor is not essential, and that frequent hitting by common or semi-armor-piercing shell may be decisive. Thus, the Japanese concentrated their fire on the *Suvoroff*, the Russian flagship, as soon as fire was opened, and five minutes later the *Suvoroff* began to suffer from the fire and changed her course four points to starboard. Twenty minutes later she left the line ablaze fore and aft, unable to withstand the fire. It is quite evident that the Russian battleships would have been easily repulsed by our 8, 10 and 12-inch seacoast guns, firing shell, and that far less punishment than these ships received would have caused them to draw off. The protection afforded by the belt armor is a maximum only when the ship is at a certain trim, which cannot always be maintained. Thus, at Tsushima, owing to the large quantities of coal on board, some of the belt of the best Russian ships were too deeply immersed. Also, a ship of great beam, if light in the water will expose her unarmored sides below the belt if rolling through a small angle. The destruction of funnels proved disastrous in the case of the *Suvoroff*, and one of the most vulnerable features of a modern battleship is its fire-control system. The experiments on the *Hero* were ordered not so much to ascertain the effect of fire on armor as upon the highly developed fire-control system. For this purpose the British Admiralty placed a complete installation of the standard type on the *Hero*. A 6-inch shell burst in the fire-control top on the mast and carried away the range-finding station early in the simulated combat; but most significant and important of all a splinter of shell, not a direct hit, had passed through the mast, and although it still stood, every fire-control wire in it had been severed. This was accomplished two minutes after fire was opened. The *bursting charge* should be of high explosive. Most of the powers have developed a satisfactory high explosive, having the most necessary qualities of stability, safety and insensibility. The development of a stable and safe delay-action detonating fuse, cannot be said to be in a satis-

factory state. Results of recent proving-ground tests, however, indicate that this type of fuse will be perfected.

5. *The policy* followed by the different powers is of interest: The latest ships in the *British naval service* are being provided with ammunition as follows, taking the *Dreadnought* as an example:

12-inch guns, 80 shots per gun, consisting of:

16 A.P. shell.
48 Common shell.
16 H.E. shell.

The latest ships in the *French naval service* are being provided, taking the 30.5 cm. gun as an example, with 60 shots per gun, consisting of 20 A.P. shell and 40 common shell.

The more recent ships of the *United States naval service* are provided with ammunition as follows:

12-inch guns, 60 shot per gun, consisting of:

24 A.P. shell.
36 Common steel shell.

7 and 8-inch guns, 100 shots per gun, consisting of:

30 A.P. shell.
70 Common steel shell.

The *British seacoast defenses* are provided with A.P. shell and H.E. shell for guns of 6-inch and larger caliber. The A.P. shell has a black-powder burster, and is fitted with a delay-action fuse. The H.E. shell carries a bursting charge of Lyddite, and is fitted with a non-delay-action fuse. The proportion of A.P. shell to H.E. shell is 6:4.

In our service, the present policy with special reference to the primary armament is to provide A.P. shot and A.P. shell for direct-fire guns, in the proportion of 2:4, and to provide D.P. shell of two weights for the 12-inch mortar; the number of shots in each case being based on the rate of sustained fire for the piece, for an action of one hour's duration, all guns being engaged.

6. *The policy that we would recommend,* involves the adoption of two new types of projectiles, resulting in a simple and less complicated ammunition supply.

(1) The adoption of a universal armor-piercing projectile, capable of perforating armor equal to its caliber in thickness, and carrying a high-explosive bursting charge, equal to 4 per cent. of the projectile weight, to be detonated by a delay-action fuse. This projectile to be used at those ranges at which the perforating power of the gun is equal to the heaviest armor. Such projectiles have already been developed in this country and abroad.

(2) The adoption of a common steel shell, carrying a bursting charge of high explosive, equal to 10 or 12 per cent. of the projectile weight, to be detonated by a non-delay-action fuse. Such projectiles have already been developed in this country and abroad. This type is cheap when compared with A.P. shell and are easy of manufacture. This projectile to be used at mid and long ranges (5,000 to 12,000 yards), and by certain batteries at all ranges.

These two types should be supplied generally in proportion of one A.P. projectile to four common shell, but this proportion should not apply to all batteries. The character of projectiles to be supplied each battery should be carefully worked out and be based upon local conditions entirely. Thus, certain batteries, due to their tactical location and the hydrographic features of their field of fire, will only be able to participate in long-range actions, or to secure unfavorable angles of impact, and hence should be supplied with a larger proportion of common shell. Also, a battery so located as to be able to enter the action at mid-range and to secure favorable angles of impact, should be supplied with a larger proportion of A.P. projectiles. In addition, certain batteries of intermediate guns should be supplied entirely with common shell, with a view of attacking the vulnerable unarmored parts of the target.

7. *Best Method of Attack* (*based on our present policy*).—However heavily armored a vessel may be, by far the larger area is open to attack of shell, and in this respect the more recent battleships are vulnerable. The special function of our A.P. shell, carrying a bursting charge of high explosive, is to attack these parts of the target. Shell have little perforating

power, and their destructive effect is due to the splinters of shell and debris of the structure, which have a great effect against the personnel, but little on material. The bursting charge of high explosive has both moral and physical effect against personnel between decks and may cause serious destruction to material in addition. If the bursting charge is black powder the incendiary effect is valuable. Our D.P. shell is similar in every respect, except in some minor dimensions, to our A.P. shell. The special function of this projectile, which is used in the 12-inch mortar, is to attack the ship in its most vulnerable part, through the decks. This projectile if provided with a slight delay-action fuse should be most efficient in destructive effect.

The function of our A.P. shot, carrying a small charge of high explosive is to attack material, by perforating the main belt, barbettes, turrets, and thus render the vessel unmanageable; injury to machinery, guns, steering gear, etc.

We have seen, however, that the use of such projectiles is limited to those shorter ranges at which the perforating power of the gun and projectile are equal to the thickest armor.

8. *The best aiming point* is the base of the foremast. In all recent vessels some of the most important guns, the conning tower, communications, central station, etc., are located near the base of the foremast, and to a certain extent this space is also crowded with men operating the guns, ammunition service, machinery, etc. It may appear at first thought that it is entirely impracticable to concentrate the fire on any particular part of the ship. In the accompanying figure the 30 by 60 foot target used in target practice is placed on a recent ship.

From a study of the figure it will be seen that the selection of an aiming point is not only practicable, but should be insisted upon. In exceptional cases of belt attack with A.P. shot it is best to direct the fire well forward, as even if not sunk, a ship with forward compartments filled would be down by the head and difficult to manage, and therefore an easy target.

9. *Fire Tactics.*—In conclusion, the fire tactics applicable during the different stages of an attack may be summed up as follows:

(1) It is desirable to disable the enemy as far away as possible, hence fire should be opened as soon as the target enters the First Defense Zone (12,000 to 8,000 yards). In this zone only a limited number of hits will be made and the perforation of armor is not to be expected. Hence the projectile to be used should be that which will cause the most destructive effect.

In this zone the destructive effect of a projectile depends more on the bursting charge than on power to perforate. Therefore it would appear that shell should be exclusively used.

(2) The greatest amount of damage may be expected to be done to the enemy in the Second Defense Zone (8,000 to 4,000 yards). The fire of the batteries will be more fully developed and should suffer but little from the secondary armament of the enemy's ships. More hits may be expected in this zone and some of the lightly armored parts may be perforated. Hence some A.P. shot may be used, but the use of such projectiles should be strictly limited to the most powerful batteries, so located as to obtain favorable angles of impact and ranges.

(3) In the Third Defense Zone (4,000 to 1,000 yards), the probability of hitting is great and perforation of main armor by guns of the primary armament may be expected. Hence, A.P. shot, if available, should be used by all guns of the primary armament. Angles of impact will be more favorable, and near the entrance proper the target will expose her broadside to something like normal impact. Guns of the intermediate and secondary armament should continue the use of shell.

(4) The 12-inch mortar, being provided with one type of projectile, will use the D.P. shell at all ranges. Fire from this armament should open as early as practicable.

PROPELLING MACHINERY OF WARSHIPS.*

By Engineer Vice-Admiral H. J. Oram, C.B., Engineer-in-Chief of the Fleet, President of the Junior Institution of Engineers.

The following address by the Engineer in Chief of the British Navy is of special value in giving correct insight into the present engineering development in the English Navy, as a forecast of probable future developments in that service and as indicating along what lines improvements or modifications may be expected.

The attitude assumed may be considered by enthusiastic promoters of some of the new and as yet untested schemes and devices as being somewhat conservative. However, the fleet of the nation's fighters must, above all, be reliable, and for this reason every device and system materially affecting the motive and maneuvering power must be thoroughly practicable, tried out and tested to meet service requirements before real dependence can be placed upon it and before it can be considered fit in all respects.

A new device, before it can be introduced, must give promise of being able to surpass the existing device to a considerable extent, and there must be reliable data at hand to show that the results promised have an overwhelming probability of being accomplished, and the theory must be backed in its essential points by the results of practical demonstration.

The address does not appear to show that any very startling or unthought-of developments are underway, but, as far as that is concerned, new developments in the British Navy are often kept well under cover till they have been thoroughly developed.—H. C. D.

*Extracts from Presidential address delivered at the inaugural meeting, Nov. 16, 1909.

FORCED LUBRICATION.

Previous to its adoption in large ships forced lubrication had been applied to the main engines of several torpedo-boat destroyers with great success, and although some extra weight was involved by these designs, this was compensated for by less liability to accident and saving in friction of engine bearings. As a testimony to its success for such purposes it may be stated that on examining the main engines of one of the first destroyers fitted with this system, after they had been running 20,000 miles it was found that the original tool marks in the white-metal bearings were still visible, and no measurable wear had taken place. This experience with the smaller ships led to the introduction of forced lubrication for the crank pins, eccentrics and main bearings of the engines of the larger ships previously named. It was found in practice to be a great improvement, and eliminated entirely the risk of accident due to hot bearings caused by ineffective oil supply, which was the principal trouble of the marine engineer at that time.

The method adopted in the destroyers was to enclose the whole of the working parts, including the crosshead bearings of the connecting rods, in an oiltight casing, packed glands being provided in the casing for the piston and slide rods to work through. Observation windows and electric lights were fitted. The oil was supplied by pumps worked by eccentrics on the crank shaft, taking their supplies through strainers, and delivering into the hollow crank shaft and weigh shaft, pressure gauges being fitted on the delivery pipes. Holes were drilled approximately radially at each main bearing, eccentric and crank pin, and through connecting rods, &c., to supply the oil to the various surfaces.

In the larger ships with reciprocating engines, pumps, separate from the main engines, were fitted for circulating the oil, the delivery pipes being provided in duplicate. The oil supply to them was first led to the main bearings, and thence through the shafting to the others, and in these ships it was not considered necessary to apply the forced lubrication to the cross-

head pins or link gear. With a view to separating any water which might mix with the lubricating oil, owing to drainage from side and piston-rod glands, an oil-settling tank was fitted in each engine room. Although the adoption of forced lubrication on such an extensive scale as for reciprocating engines is not required in the case of turbine machinery, yet in this case as well, the principal bearings—viz: the turbine spindle and propeller-shaft bearings—are provided with this system.

Forced lubrication had previously been so successful in overcoming bearing troubles in electric-light engines that it has now been applied to the great majority of auxiliary engines, with satisfactory results as regards behavior of bearing surfaces, and many of these engines are consequently now run at a high speed of revolution. One difficulty arose in the application of forced lubrication to reciprocating engines, viz: the constant splashing of the oil on the piston rods, and the entry of this oil on the rods into the cylinder on the in-stroke caused a considerable amount of oil to pass with the exhaust steam to the condensers, and from these to the feed water, and finally to the boilers.

Oil in boilers is a fruitful cause of overheating and other similar troubles, and it was necessary to deal with this phase of the question, so that modifications of design and superior glands have been fitted in many cases, and this, with improvements to grease filters, has so minimized this difficulty that, with proper attention to the fittings provided, it becomes unimportant. The adoption of turbines has been a great improvement in this respect, as the quantity of oil passing to the condensers with the steam from them is practically nil, and the grease filters in these cases have only to deal with the grease from the auxiliary engines.

The reciprocating engine in its latest form in the Navy, fitted with forced lubrication, and with its moving parts designed so as practically to balance the inertia forces, embodying improvements that had been evolved during many years, had reached a high degree of mechanical excellence, and, omitting imperfections inherent to the type, the only important

obvious feature in which it was wanting as regards fittings was the ability to use superheated steam, and secure the fuel economy which is well known to result therefrom.

It was only in 1904 that superheaters were again ordered in His Majesty's Service, and on a small scale. Six Babcock & Wilcox water-tube boilers of the total of twenty-one in H.M.S. *Britannia,* a battleship of 18,000 indicated horsepower, were fitted with them in order to obtain experience as to the efficiency of superheating and the durability of the superheaters themselves, under the greatly altered condition of boiler construction and treatment as compared with the conditions obtaining when the superheaters were abandoned. The construction of boilers had considerably improved since the early days, and higher steam pressures and greater rates of expansion of the steam were employed. Boilers are now worked with practically fresh water, the cylinders and slide valves were vertical instead of horizontal, the workmanship and details of these parts had improved, metallic packings were used for piston and slide-rod glands instead of the more perishable earlier packings, and the steam-pipes are, and mountings can be, of steel. All these details of design are conducive to the better practical application of superheating than was the case forty or fifty years ago.

Although the *Britannia's* engines embodied the improvements in design just mentioned, it was thought desirable to proceed cautiously with the fitting of superheaters owing to possible increased wear of rubbing surfaces of the engines, and for this reason only a portion of the boilers were so fitted. It was not thought that the early troubles with the superheaters themselves would be repeated, although it was not certain that all difficulties with them would be eliminated.

The actual alterations made in these six boilers during the design stage, as compared with the remainder, was that one-eighth of the tube surface was removed and the same amount of superheater surface added, the gases passing over the superheater surface after they had traversed the first third of the heating surface. The general results were as follow: Two

thirty-hour trials at one-fifth full power were run, six superheater boilers being used in one case and six ordinary boilers in the other; the steam used per indicated horsepower per hour for all purposes was 18.19 pounds with superheaters, and 21.02 pounds without superheaters, showing an economy of 13½ per cent. in water consumption due to superheating. The actual amount of superheat obtained was 93 degrees F. at the boilers, with a gauge pressure of 200 pounds, which fell to 84 degrees F. in the high-pressure receiver, with a gauge pressure of 86 pounds per square inch.

The high-pressure cylinder liners are of forged steel, the piston rings being two narrow phosphor-bronze restrained rings, and after the trial with superheated steam these cylinder liners showed signs of "dragging."

Further trials were then made with superheat of 30 degrees and 61 degrees F. After the trial with 61 degrees F. superheat there was an increase in the "dragging" action noticed previously, the surfaces were very "brassy," and flecks of metal from the piston rings were embedded in the cylinder walls, while with the 30 degrees F. superheat the condition of cylinder liners improved.

The results of these trials led to the conclusion that, as then fitted, it was not advisable to work the engines with steam superheated beyond 30 degrees F., and to admit of their always being so worked a connecting pipe was added, so that a mixture of superheated and ordinary steam could be used as desired. In this condition, however, the steam can hardly be much more than dried, and no marked economy could be expected on service.*

Internal lubrication was not used on any of these trials, in accordance with the established practice of the Admiralty; but if it had been used, it would probably largely have reduced the action described, but with the present means of feed-water filtration the admission of such lubricant is objectionable.

*It does not appear that the use of phospor-bronze H.P. piston ring for superheated steam was the wisest course. Cast-iron rings working on cast-iron liners have proven entirely satisfactory with superheated steam of 500 degrees F.—H. C. D.

There is no doubt that, had the reciprocating engine remained the standard for marine purposes in the Navy, this matter would have been pursued further, and the difficulties would have been dealt with, and no doubt overcome, in future ships by improved fittings and materials.

It is more important to us, as regards the future, to notice the behavior of the superheaters themselves after about three and a half years' work, with the view of determining whether the troubles of years ago are likely to be repeated, and there does not appear to be any such likelihood with the degree of superheating attempted, which was not great. The ship has been at work since 1906, and the superheater boilers have been used equally with the remaining ones, and no defects have been reported due to the superheaters. Periodical reports of examination of the condition of the superheater tubes have been made, and a few are removed and cut up for examination at intervals. The most recent report states that the interior of the superheater tubes was coated with a rusty deposit, and active pitting is taking place, which is general. In the worst cases, however, the pitting does not extend beyond 0.04 inch, the original thickness of the tubes being 0.144 inch, and the tubes are good for several years' further service.

The deposit inside the tubes consisted of a mixture of ferrous and ferric oxides, with quite small amounts of chlorine, carbonic anhydride and lime. The latter constituents form an indication of the moisture carried to the superheater with the steam. They amounted, on the average, to: Carbonic anhydride, 0.32; chlorine, 0.1; and lime 0.52 per cent. respectively.

SUPERHEATED STEAM IN TURBINES.

When we consider the use of superheated steam in turbine machinery, a different set of conditions and requirements was met with, and although no difficulty was to be anticipated as regards wear of parts due to dryness of steam, the alteration of relative position of parts, consequent on differences of expansion due to variations of temperatures, is materially in-

creased by superheating. This increased difference of expansion between the various parts of the turbine casing and drum from the cold condition to that when at work due to temperature renders larger clearances of parts necessary; and there are further objections, due to the effect of possible variations of superheat. The use of superheated steam, therefore, in turbines containing close-fitting parts, did not seem to warrant the extra risks involved by it, so that the fitting of superheaters received a check. Besides this, the results being obtained in the Parsons turbines with ordinary saturated steam were extremely good.

At present, too, as compared with the early days of low-pressure steam, the difficulties of obtaining adequate superheating are greater. Compared with the practice of forty years ago, the temperature of the steam itself has been considerably raised by successive increases of pressure, while at the same time, by improvement in boiler efficiency, there has been a continued reduction of the temperature of the gases discharged from the boiler. The difference between these two temperatures which is now available for superheating is considerably less than it used to be. There was no difficulty in the old days in obtaining, within reasonable limits, as much superheat as was desired from the gases finally leaving the boiler without any modifications to the latter, which would make it less efficient, and the heat abstracted was so much clear gain. At present, however, this is not possible; the gases are so reduced in temperature by their passage over the normal heating surface that unless special arrangements are made the excess over the steam temperature is too small to be of much use for superheating.

With the ordinary type of Yarrow or Babcock & Wilcox boiler used in the British Navy, there is therefore difficulty in obtaining any considerable superheat by simply placing the superheater among the gases which have just left the boiler. Superheating with the present-day boilers therefore requires some alteration in boiler proportions or arrangement, in order

to obtain such a difference between the temperature of saturated steam and that of the funnel gases in contact with the superheater as will secure sufficient superheating. This modification somewhat reduces their efficiency as boilers, and the resulting gain, due to the use of superheat, is the difference between these two items, which have each to be examined so as to be certain that there is sufficient gain on balance. An alternative for large ships has been suggested in the fitting of separately oil-fired superheaters, which would remove the objection noted previously to modifications in design of boilers, the extra space and weight of the separately-fired superheaters being compensated for by reductions in the number of ordinary boilers.

At the other end of the temperature scale very important considerations occur as regards the temperature and vacuum when the steam leaves the turbine, for it is well known how eminently suitable the marine steam turbine is for the economical utilization of very low pressures. It has a great superiority over the reciprocating engine in this respect, and the reasons for this may be usefully referred to. The amount of work obtained from a given quantity of steam depends principally on its final volume after expansion while doing work, and this final volume depends on its velocity and the sectional area available for the flow of steam. In the reciprocating engine its velocity when doing work cannot exceed that of the piston, which is limited by practical considerations quite independent of those of steam efficiency, and its sectional area—*i.e.*, the area of low-pressure cylinder—cannot be increased much, for in this case we should have enormous and impracticable cylinders and large losses due to friction, condensation, and in other directions. In the turbine there is no similar limit to the steam velocity, and the steam can be expanded to within about 1 inch of the condenser vacuum, so that as the steam annulus at the low-pressure end is large, and the velocity of the steam high, an enormous increase in volume is obtained during expansion while in contact with the turbine blades.

RANGE OF EXPANSION IN RECIPROCATING AND TURBINE ENGINES.

In naval reciprocating engines it is well known that, owing to considerations of weight and space, a large expansion was not given at full power; in other words, economy at full power had to be sacrificed to other requirements in the warship compromise. Taking a number of representative reciprocating engines for our more recent warships with a fairly high steam pressure—viz: about 220 pounds per square inch by gauge in high-pressure receiver—the average pressure at the completion of the expansion at full power is 12½ pounds absolute, with a back pressure, whilst the cylinder is open to the exhaust of 6.7 pounds absolute, the average vacuum being 26 inches, or, say, an absolute pressure of 2 pounds per square inch.

Further, whilst the average back pressure in the low-pressure cylinder was about 6.7 pounds absolute, or 16½ inches vacuum, the expansion only extends whilst driving the piston to about 12½ pounds absolute, or 5-inch vacuum, so that a considerable percentage of the energy in the steam which could be obtained by gradual expansion from 12½ pounds to 6.7 pounds while doing work, is also lost.

In the turbine, however, instead of these comparatively high pressures at the termination of expansion, we have the fact that the steam does useful expansive work to about 27 inches vacuum at the turbine end, with a vacuum of 28 inches in the condenser, the necessary volume being obtained as indicated previously. At the other end of the scale the average steam pressure at the turbines has only been about 140 pounds in recent examples.

Assuming adiabatic expansion, the following table shows the energy available in thermal units per 1 pound of steam when expanding between various pressures, and the comparatively large amounts obtained for small drops of pressure at low pressure:

Expanding from 235 pounds absolute to 16½ inches vacuum, 249 thermal units.

Expanding from 16½ inches to 20 inches vacuum, 16.2 thermal units.

Expanding from 20 inches to 22 inches vacuum, 11.8 thermal units.

Expanding from 22 inches to 24 inches vacuum, 15.1 thermal units.

Expanding from 24 inches to 26 inches vacuum, 21.4 thermal units.

Expanding from 26 inches to 28 inches vacuum, 31.9 thermal units.

It will be seen that the B.T.U.'s given up by the steam expanding adiabatically from a vacuum of 16½ inches to one of 28 inches is 38.7 per cent. of the B.T.U.'s available from an adiabatic expansion from 235 pounds absolute to 16½ inches vacuum, showing the great gain by the economical expansion to low pressures, and explaining the superior economy of the turbine.

THE EFFICIENCY OF THE TURBINE.

When it was decided in the *Dreadnought* to supersede the ordinary reciprocating engine by the Parsons marine steam turbine, careful analyses of the results obtained in previous vessels indicated that an increased economy would be realized over the average type of reciprocating engine of that period, to an extent which would enable the S.H.P. of the turbine to be the same as the indicated horsepower of the reciprocating engine, while using about 15 per cent. less steam.

The machinery of the *Dreadnought* was designed on these lines—*i. e.*, a reduction of about 15 per cent. was made in the usual boiler proportions, and the trial results confirmed the action taken, the consumption of steam at full power for turbines only being 13.48 pounds per S.H.P. per hour, as carefully ascertained from large measuring tanks. The initial pressure at the turbines was 164 pounds per square inch by gauge.

The turbines of subsequent ships have been gradually improved with experience, and although the general construction remains the same, clearances and other adjustments required

are by experience better known, and improved fittings and measuring appliances enable the dummy clearances, on which steam leakage very much depends, to be reduced, so that subsequent results have been even more satisfactory than those of the *Dreadnought.* The average steam consumption of the three battleships succeeding this ship was 13.01 pounds per S.H.P. per hour for turbines only, with an average gauge pressure of 147 pounds per square inch on the high-pressure turbine.

The turbine engines of the *Indomitable* class of 41,000 S.H.P. were much greater in size and of slower revolutions than those of the battleships, and the effect of this on the efficiency of the turbines could not be quite foreseen, so rather a conservative estimate was made of the probable steam economy; and, though the boilers were made a little smaller proportionately (about 4 per cent.) than in the *Dreadnought,* trials of these ships showed that the steam consumption of the turbines was lower than expected. The results were superior to anything that could have been predicted with confidence at the time, as the average consumption of steam at full power for turbines only was 12.03 pounds per S.H.P., with the average gauge pressure 123 pounds per square inch at the high-pressure turbine.

CRUISING TURBINES.

All our original turbine ships were fitted with cruising turbines in addition to those for use at the higher powers. The *Dreadnought,* for example, has four turbines on each side of the ship, counting as one turbine the low-pressure and low-pressure astern, which are in one casing. Experience, however, has shown us that there are certain inconveniences attending the use of cruising turbines. They are economical, but being very often not in use, they are apt to be neglected, and not to receive the attention they require, and the few accidents that have occurred with turbines have practically all occurred in the cruising turbines.

The question arose as to whether the increased economy due

to their use is worth the extra complication, cost and liability to injury, and the conclusion is arrived at, at least in the case of single cruising turbines, that the balance of advantage and disadvantage is adverse to them, and they have not been fitted in recent warships. This view is the more readily taken as the alternative recommended by Mr. Parsons appears to give us most of what we require without multiplication of turbines. This alternative is considerably to increase the expansion allowed in the main turbines at high powers, provision being made to obtain the maximum power by means of by-pass arrangements. This ensures greater economy at low powers than was hitherto obtainable with the main turbines, and by this system we gain simplicity and also a reasonable economy at such low powers.

In certain classes of ships, however, where the radius of action at low powers is exceedingly important owing to their limited coal or oil storage, as in torpedo-boat destroyers, two cruising turbines in series are fitted. In a typical example of this sort there are five turbines fitted on three shafts, again counting the low-pressure and low-pressure astern turbine as one. In this case the gain by the use of the cruising turbines in series is much greater, and is considered to make up for the extra cost, weight and complication involved, and they are being retained.

THE STEAM CONSUMPTION OF AUXILIARY MACHINERY.

A few remarks on the consumption of steam in auxiliary engines will now be made. These engines, except in electric-light and hydraulic pumping engines, are usually of the single-cylinder type, and their efficiency is therefore small. An average consumption of steam, when exhausting to the condenser direct, of the reciprocating electric-light engines which are compound, is about 28 pounds per indicated horsepower. For main circulating engines the consumption is about 45 pounds per indicated horsepower, whereas for pumps, steering engines, capstan engines and other engines, in which there is little or no

expansion of the steam, the average consumption is about 60 pounds per indicated horsepower. These are trial results, and it must be remembered that with wear on service these figures would be increased.

This being so, it is obviously desirable to abstract some of the energy existing in the exhaust steam from such auxiliary engines. This was accomplished by working them on what is called the "closed exhaust," which simply means closing the direct connection of the auxiliary exhaust pipe to the condenser and passing the exhaust steam to the main engines at a position where the pressure is suitable, or through the coils of an evaporator. Relief valves loaded to about 25 pounds per square inch are fitted, and allow any surplus exhaust steam to pass direct to the condenser.

Considerable economies are obtained by the use of "closed exhaust," especially when used in the evaporators, where 1 pound of gained steam can be obtained from about $1\frac{1}{4}$ pounds of exhaust steam. When not required in the evaporators, the steam in the case of reciprocating engines is admitted to the low-pressure receivers, and some gain is effected thereby. With the adoption of turbines, however, with their high efficiency in the use of steam of low pressures, the gain from the use of auxiliary exhaust in the main engines has been considerably increased, and this constitutes a very important advantage for turbine machinery as regards warships. Taking a considerable number of ships, it is found that by this means the consumption of steam for auxiliary engines is reduced by half, thus: If the consumption of steam for the main turbines and for auxiliary engines per S.H.P. when not using the auxiliary exhaust are represented by a and b respectively—*i.e.*, a total for all purposes of $a + b$, it is found that the total consumption of steam for all purposes when using the closed exhaust on the turbines will be approximately $a + \frac{b}{2}$. It is interesting to note that the combination of reciprocating engines and turbines recently fitted in mercantile ships is similar to the closed exhaust system working on the low-pressure turbine.

COMBINATION OF RECIPROCATING AND TURBINE ENGINES.

Economy at low powers is, in a warship, of more importance than at high powers, and as low powers can only be obtained in the turbine by considerable reduction of initial pressure, there being nothing corresponding to the variation of cut-off with the same initial pressure as in the reciprocating engine, the combination of reciprocating engines using high pressures exhausting into a low-pressure turbine appears economically attractive for the development of the lower powers in warships.

Let us consider the ordinary four-shaft arrangement in battleships, but with a reciprocating engine fitted on each of the outer shafts, and a low-pressure turbine on each of the inner shafts, the total horsepower at full power being about equally divided over the four shafts, the principal object of this arrangement being to obtain the best economy at about one-tenth full power.

Under full-power conditions the initial pressure at the low-pressure turbines would be about 22 pounds absolute, and the exhaust pressure at the reciprocating engines about 2 pounds above this figure. The results of investigation show that at full speed the cruising radius would be approximately the same as that with an all-turbine arrangement. At one-tenth power the cruising radius of the combination would be about 10 per cent. better than an all-turbine arrangement with cruising turbines in "series," on the basis of a propulsive coefficient of 50 per cent.

At low speeds, however, the distribution of power over the shafts of the combination would be very unequal, and it is difficult to estimate the propulsive coefficient that might be obtained. This, together with the extra consumption of oil, &c., and the inherent practical disadvantages of reciprocating engines as compared with turbines, appear to more than outweigh the possible additional economy which might be obtained at the lower powers with the combination.

The combination system generally appears suitable for the class of vessel where the designed speed falls below the limit

of applicability of an all-turbine arrangement, and from the results published of such installations it would appear that in these cases additional economy of at least 10 per cent. could be obtained over good quadruple engines, and at least 15 per cent. over ordinary triple-expansion engines.

THE ELECTRICAL PROPULSION OF SHIPS.

Referring first to what is called the electrical propulsion of ships: An electrical drive has already been adopted in some ships of small power where the prime mover is an internal-combustion engine, in order to overcome difficulties in starting and to avoid complicating these engines with reversing-gear. In one system an electrical drive is adopted for reversing or slow speeds only; at other speeds, though the dynamo and motor are coupled to the engine shaft, no electricity is generated, and the internal-combustion engine is working directly on the propeller. In this design the capacity of the dynamo and motor is limited to about one-half the power of the engine in order to save weight.

Since the introduction of turbine machinery in warships proposals to supersede turbines directly coupled to the propeller shafting, by designs where electrical machinery is interposed between the turbines and the propellers, have been made several times.

The system, as proposed, consisted of the employment of economical turbines, at very high revolutions, similar to those in use on land electric-light stations, with a dynamo and motor interposed between the turbine and shaft to reduce the speed of the latter to that which is most suitable from the propeller-efficiency point of view. The turbines are always run in one direction, and reversing can be effected by the change of the direction of the current at the motor.

The claims made for this system are that the loss due to the interposition of the dynamo and motor between the turbine and propeller shaft is more than counterbalanced by the gain due to the more economical steam consumption of high-speed tur-

bines, and that due to a higher propeller efficiency when working at more moderate speeds than is usual in ship turbine machinery.

It is also claimed that the radius of action of the ship will be increased due to an economy when working at low powers, dynamo machines being shut off altogether, as required, and the remainder worked at, or near, their economical power, and that although the high-speed turbines and electrical machines are together heavier than the direct-coupled turbines this is more than compensated for by the lesser boiler power required, consequent on the higher efficiency of the fast-running turbines and slower-running propellers, so that a complete installation for the same speed of vessel is not only more economical, but no heavier. Further, that there would be an increased economy from working the auxiliary machinery by motors driven from the main generators.

These are the main features of the system; there are various other claims made, but they are not important compared with the preceding.

The actual figures are the subject of considerable doubt and controversy, and would require much time for an adequate examination. I shall only refer generally to some of the important features.

As regards economy at full power, it must be remembered that the claims are based on figures obtained from trials of electric-light stations on shore, and it is somewhat doubtful if the same results will be obtainable when similar units are worked as a battery on board ship with the engines and boilers placed in from five to seven or even more separate watertight compartments.

It appears that the loss which results at the dynamo and motor is not less than 12 per cent., so that not more than 88 per cent. of the shaft horsepower given out by the turbine would be available as horsepower at the propeller shaft.

The actual steam consumption in pounds per S.H.P. for the propelling turbines in the three ships of the *Indomitable* class is only 12 pounds at full power, and this without superheating.

The figures guaranteed for the steam consumption of the electrical combination from experience in shore electric-light installations were given in one case worked out in detail as 12.9 pounds per brake horsepower at the propeller shaft without superheat, and 10.9 pounds with 100 degrees superheat, the vacuum being 28 inches. It will be seen, therefore, that in the particular case of the cruisers mentioned there is little margin of superiority at full power available for reduction of boiler power, even allowing for the probable improvement in economy of the electrical installation due to the larger units in these ships.

The claim to increased economy at low powers should be realized, as the generators then in use could be worked nearer their maximum output; but it must be remembered that the efficiency of the motor falls off with reduced powers, but the actual amount is doubtful. This feature of the system is the important one as regards machinery of warships. The claim as regards increased power for going astern is also justified.

As regards increased propeller efficiency, it must be remembered that the efficiency now obtained with the slower-running propellers has been the result of many years' experience. The early turbine propellers had low efficiencies; experience has enabled these to be increased, but they do not yet reach the figures obtained with the slower-running reciprocating engines. There is no doubt that as more experience with small fast-running propellers for such large powers as have only been recently introduced is obtained, the efficiency of the latter will be still further improved. In particular cases, such as in the large cruisers of the *Invincible* class, there has been no falling off in propeller efficiency of the faster-running screws from efficiencies obtained in the slow-revolution screws of older cruisers.

No doubt a large-diameter propeller assists the maneuvering power of a ship, but the diameter is frequently limited by hull considerations; and electrically-driven ships with the same number of propellers, driven slower, would in some cases be

limited to practically the same dimensions as in direct-driven installations.

In one design, in which the weights of the turbo-generator and motor were quoted, it was found that the increase of engine-room weights would be considerable, and that the total increase of machinery weights would be about 15 to 20 per cent. The gain at low powers would require to be very substantial to balance such increases. The substitution of electric motors for auxiliary steam engines would also add to weight, and such auxiliary machinery as steering engines and distilling plant would for the present require to be steam driven.

The electrical pressure proposed in the various plans has varied from 1,000 volts in the lowest to 1,750 volts in the highest, and these are all very high, from the point of view of safety, for use in H.M. ships. Also, before any system could be adopted many necessary details would require to be carefully considered, such as freedom from danger of failure due to access of water to the electrical machines or to particular parts of them, and the ventilation of the machines, as considerable heat is generated.

With the interposition of two electrical drives between the turbines and the propeller, in addition to special dynamos as exciters, it is not probable that the wear and tear would be less or that less staff would be required, but rather the reverse.

In the designs submitted to the Admiralty there has been no gain in the space occupied, but in some cases an increase.

It is essential for warship purposes that the small variations of speed required from the propellers when keeping station in a fleet should be readily obtained, and in this respect the electrical systems which have been proposed to the Admiralty are far from satisfactory. The various firms have only proposed to provide for three electrical speeds—full, half and quarter—and have suggested that if anything further were required that the speed of the generators should be regulated. This latter would detract from the economy of the system.

It is recognized that further attempts are being made to meet this condition, which is absolutely necessary as far as warships

are concerned, provision for station-keeping or the power of making gradual changes of speed throughout the whole range, both ahead and astern, being essential.

Finally, there appears to be considerable doubt as to the efficiencies of the various systems proposed, and the figures on which they are based; but, speaking generally, the conclusion seems to be that the interposition of a dynamo and motor between the engine and its work, involving when on board ship considerable liability to accident and derangement of the electrical machines, in addition to present risks, has disadvantages such that the practicability of the details should be beyond question, and the gain by its adoption must be shown to be substantial before the system could be fitted in any important vessel. This has not, I think, been done at present. We must recognize, however, the skill and ingenuity with which this problem has been dealt with, and any further developments will naturally receive careful consideration, especially the actual results from installations which are about to be fitted on a small scale. The good work done by electrical engineers on board warships at present in respect to auxiliary machinery of all kinds is too well known to need any comment from me, and we should welcome their co-operation as regards the main engines, if the resulting combination is likely to be beneficial.

THE INTERNAL-COMBUSTION ENGINE.

We now pass on to some general remarks on the use of internal-combustion engines for marine propulsion. It is a difficult matter to make exact comparisons between steam engines and internal-combustion engines suitable for large ships, as the powers of the latter installed afloat have not been large, and comparisons of units of 500 brake horsepower in internal-combustion engines and 20,000 to 40,000 brake horsepower in steam engines leaves much to be desired; it must, of course, be remembered that the steam engine is in a high stage of development, and the marine gas-engine installation not so.

A more reliable comparison at low powers can be obtained

from a comparison of steam engines and oil engines in boats, for as the weight of gas engines without the producers, &c., should be about the same per indicated horsepower as that of oil engines for the same speed of revolution and working on the same cycle, this comparison will give an idea of the position of the gas engine in comparison with the steam engine for such powers.

In the large power-driven boats carried by warships it is found that where the total weights of the heavy-oil engines and the fuel required for the same radius of action are about the same as for similar boats with steam machinery and coal, the speed obtained with the oil-motor boat has not been so great as with the steam boat. As oil fuel is assumed to be available, this could be equally used by the boiler of the steam engine, and if this be assumed, the comparison would be still worse for the internal-combustion motor. For small boats, where the air compressor and reservoir for starting can be omitted, and hand-starting gear is sufficient, the heavy-oil engines may have a slight advantage, but this would disappear if the smaller boats were similarly compared with boats fitted with oil-fired boilers.

This gives a general idea of the results obtained with oil engines which use oil of a flash point of over 200 degrees F., and shows that there is practically no gain in respect to weight in fitting heavy-oil motors, instead of steam engines, in large power boats of corresponding revolutions. There being practically little difference in the weights of four-cycle oil motors and steam engines, it follows that four-cycle gas engines with producers and cleaning plant must generally be heavier than steam engines and boilers.

The gas engine and anthracite producer plant which was fitted by Messrs. Beardmore in H.M.S. *Rattler,* after the original steam engines and boilers had been removed, is the largest marine gas-engine installation fitted in Great Britain, and we must admit the enterprize of those associated with this experiment, which has added considerable to previous knowledge of the subject. A comparison with the steam engine which was

taken out of this vessel shows that the horsepower obtained per ton with the latter was about twice that obtained with the former. This compares the steam-engine practice of 1886 with the gas-engine practice of twenty years later, but it must be remembered that the gas engine fitted in this ship was an experimental installation, and a future one would no doubt make a more favorable comparison with the steam engine.

This gas engine was not reversible, but was fitted with a reversing clutch, but it is very doubtful whether such reversing mechanisms are suitable for any considerable power, and it is considered that for such powers either the engine should be a reversible one or be fitted with electrical transmission.

There appears no doubt that gas engines can be made reliably reversible; in fact, a set of 500 brake horsepower already constructed possesses this feature, so that no difficulty would be apprehended on this account. If a reversible engine is, however, fitted, a very large compressed-air service is necessary for this purpose, and the capacity of this plant would be a serious factor in a large installation.

There would also be a considerable amount of experimental work to be done with the auxiliaries, such as distilling machinery and steering engines, if this type of engine were fitted, so that generally the combination of non-reversible gas engines with electrical transmission is the most promising direction for the utilization of this system.

As regards oil engines, it is no doubt correct that about 17 S.H.P. per ton of machinery can be obtained with special designs for using heavy oil. Each engine, however, is small, and its power cannot be increased very much without decreasing revolutions and increasing the weight for a given horsepower, so that the very large powers required for marine propulsion could only be obtained by multiplication of small units. One could imagine what a very large number of such reciprocating oil engines, each having its valves, cams, &c., would be like if fitted to make up the full power of a warship.

In a paper published a short time ago the figure of 11.6 brake horsepower per ton has been given as the probable weight

of gas engines and producers if fitted for marine propulsion. The tenor of this paper was entirely favorable to the adoption of the gas engine in lieu of the steam engine, so that if this figure errs at all it would probably be rather overestimated. Now the steam-turbine installation of the *Indomitable* actually developed 13.8 S.H.P. per ton, and the *Bellerophon* 13 S.H.P. per ton. The weight of fuel carried for same radius of action must, however, also be taken account of, and the superior economy of the internal-combustion engine is the important factor which causes these engines to be considered for marine purposes. Generally, it may be assumed that the fuel consumption in a steam-turbine installation is approximately about 50 per cent. greater than in an internal-combustion engine installation using similar fuels, and in considering the weights and spaces occupied by different types of machinery, a proper comparison must include the weights and spaces occupied by the fuel, and if this were done, the internal-combustion engine installation would compare more favorably with the steam-turbine installations.

Taking a steamship with turbines of 20,000 S.H.P. and carrying 3,000 tons of coal, and assuming that the coal consumption per S.H.P. per hour is 53 per cent. more in the steamship than in a ship fitted with gas engines and producers, and assuming that the consumption at low powers would be in the same proportion, then for the same radius of action in a gas-engine ship, with the same total weight of fuel and machinery, 8 horsepower per ton of machinery would be sufficient in the gas engine for equality of total weight.

As regards space, there is no doubt our present steam-turbine installations take up a good deal less space than any gas engine, and probably less than any oil-engine installation which it is possible at present to fit. It has been stated that about 0.32 square foot of floor space per brake horsepower would be required for oil engines in a large installation. The corresponding figure in a large turbine installation was 0.28, and this figure would decrease as the power of the turbine installation increased.

In these comparisons of weight, space and fuel large ship practice has been taken. For smaller cruisers, in which a higher speed of revolution is used, and where the boilers are worked with a higher air pressure to obtain the full power, the weight and space for steam machinery would be very considerably less than indicated above, and, further, it is not unreasonable to assume that for future large installations of turbines the weight and space will decrease.

The most important consideration is, however, the practicability of obtaining reliable large gas-engine installations and producers for use in ships. Any difficulties with the gas engine itself would, up to certain sizes, I think, be overcome, if a really satisfactory gas producer were available for marine use. This cannot be said to be the case. The use of bituminous or semi-bituminous coal is, under present circumstances, absolutely necessary, taking into consideration that anthracite can only be obtained at a few ports outside this country.

A producer using bituminous fuel and of a type suitable for marine purposes, has not yet reached a practicable stage, and until such a producer has been designed, and has proved to be thoroughly reliable after trial, the adoption of gas engines on any important scale will be prevented. The invention of a satisfactory producer would give a considerable impetus to the consideration of internal-combustion engines for ship use.

The introduction of the steam turbine has had the effect of retarding the development of the marine gas engine, for while comparison with the steam reciprocating engine was not favorable for the reciprocating gas engine, comparison of the latter with the steam turbine is worse. To engineers who have realized by practical experience the convenience and simplicity of the steam turbine, the superior merits of the internal-combustion engine, apart from the question of the producer, must be proved to be very great if the indirect crank drive, in connection with all the cams, tappets, &c., required, is again to be introduced.

There is also the difficulty with gas-engine installations that the gases are poisonous, and this would require careful con-

sideration in the arrangements for an engine room of a warship, and suction producers would, as far as this is concerned, be preferable to pressure producers. Also, after stopping or working slow for some time, it is difficult to obtain sufficiently good gas to enable a large increase of speed to be readily obtained.

An attendant feature of both gas and oil engines from a naval point of view is that funnels are often stated as not being necessary. The producer gases must, however, be got rid of, and although it may be possible to arrange to eject the gases under water when the engine is at work, when the engines are stopped an under-water discharge would not be possible in the gas engine, and some fittings corresponding to funnels must be provided.

The foregoing is only intended to be a brief review of the various systems which compete with the steam engine for marine propulsion; an exhaustive examination would occupy more than the Institution can spare me, but it indicates, in my opinion, that these rivals of the steam engine have not yet advanced much, and the steam engine holds the field for the present. Whether the gas turbine, which would be a more serious competitor than any of the foregoing, is to be successfully developed remains to be seen.

LIQUID FUEL.

Proceeding now to another subject, one of the features in the recent developments of warship machinery is the adoption of liquid fuel, either wholly or partially, for the production of steam. Its principal advantages for this purpose were fully recognized many years ago, and, as compared with coal, comprises: Greater calorific value for a given weight or space occupied by the fuels; quickness and cleanliness of transport to vessels, and much less labor in transporting and using; greater control over its supply to the furnaces and cleanliness of stokeholds; considerably less fouling of boiler tubes when under way, and no cleaning of fires necessary.

The difficulties attending its use some years ago were, however, very great, as the methods for burning it did not allow of a fair proportion of its calorific value being realized. When burnt at the rate required for Navy boilers, less than 50 per cent. of its calorific value could be obtained for some years, and dense smoke was produced. The risk of danger from explosive gases given off by the fuel was great; its great cost and limited supply were also disadvantageous.

The risk of explosion from volatile gases was reduced to a minimum in H.M. ships by the adoption of a sufficiently high flash point. The flash point first adopted by the Admiralty was 280 degrees F., but this has now been reduced to 200 degrees (Abel close test). This is higher than that allowed by the Board of Trade and Lloyd's Registry, and although the desirability of an identical flash point between the Admiralty requirements and that of other users in this country is recognized, it is considered that greater precautions are necessarily in warships, as the spaces on board are more confined and more liable to a higher temperature than those in vessels of the mercantile marine, and the effect of an explosion would be more disastrous under certain circumstances.

In determining the safe flash point, the fact must be borne in mind that vapor is slowly distilled off from the oil at temperatures lower than that of the flash point; for example, when the flash point is 150 degrees F., inflammable vapor is gradually evolved at 100 degrees F. As an illustration of the high temperatures to which certain spaces in warships may be liable, it may be stated that in the case of one destroyer the temperature in the bilges under the boilers, when steaming at full power, was found to be 158 degrees F., and although no oil tanks may be adjacent, it is quite possible, and even probable, that occasional leaks may occur from the burners, which may find its way to the bilges.

The other difficulties could not be so easily met, but experiments extending over some years were made by the Admiralty, both ashore and afloat, with steam and compressed-air spraying, and many devices, obtained both outside and inside the

Navy, were tried, but none of these proved entirely satisfactory. Both steam and compressed air as spraying agents have important and well-known disadvantages.

Efforts were, in consequence, made to develop a system free from these objections, and in 1903 a special burner was devised, by which the pressure produced by the oil pump was made to effectively atomize the oil. The trials on shore of these appliances eventually resulted in satisfactorily maintaining the full power of the boiler without the production of undue smoke, and at the same time realizing the satisfactory proportion of about 76 per cent. of the calorific value of the oil. This system was then fitted to two destroyers, and the trials proved very satisfactory. In consequence of these results, this system is now generally fitted, and the continued satisfactory results obtained have gradually led to an extended adoption of oil-fuel burning through the Service.

The full advantages to be derived from the use of oil fuel can only be obtained when it is used by itself; but this course cannot be taken in all the ships of the Fleet, on account of the greater cost and limited supply of oil fuel as compared with coal. Oil as the sole fuel is therefore confined to the smaller vessels, which obtain substantial tactical advantages by its use. Larger vessels, such as battleships and cruisers, only use oil in conjunction with coal, and in these cases the occasions on which oil is used are of a special character.

The influence of the limited quantity of oil fuel available can be realized when it is mentioned that the total amount of crude oil obtained each year from all sources throughout the world is only about 3 to 4 per cent. of the yearly production of coal, and that only a proportion of this is available for fuel. The quantity of oil which can be obtained from British possessions, or from countries under British influence, is comparatively small; but the production from them is gradually increasing, and it is probable that much greater quantities will be forthcoming from such sources in the not distant future.—"Engineering."

THE FÖTTINGER HYDRAULIC TRANSMITTER SPEED REDUCTION AND REVERSING GEAR FOR TURBINES.

The following description of the Föttinger Hydraulic Transmitter is extracted from a description given in "Engineering," November 5, 1909. This apparatus is designed to accomplish much the same purpose as the Melville & Macalpine Reduction Gear, a description of which, also taken from "Engineering," appeared in the November, 1909, number of the JOURNAL. The Föttinger Transmitter, however, accomplishes another office, that of reversing, and thus obviates the need for reversing turbines and secures almost full backing power to marine turbines, when fitted in conjunction with "Transmitter."

The development and test of these two reduction gears opens up an entirely new field to marine engineering and gives the question of turbine drive a new phase, which, as it develops, may serve to produce a marked change in marine turbine practice.

This gear was installed by the Vulcan Company on the steamer *Transformator,* of 76 tons displacement, 500 horsepower and 12.5 knots speed.

The system is urged as giving advantages in weight, space and economy over the regular turbine installation on naval vessels. On a torpedo-boat destroyer it is estimated that there will be a saving of 6.7 per cent. in floor space and 10 per cent. in weight over an ordinary installation of Curtis turbines. Here it is proposed to operate the turbine at 2,200 revolutions per minute, and to reduce this to 480 on propeller shaft. For a battleship of 30,000 S.H.P. it is claimed that there would be a saving of 23 per cent. in space and 17 per cent. in weight.

As the efficiency of the transmitter is about 80-85 per cent., it does not appear that any very marked gain in economy will

result. The high-speed turbine will probably be not more than 10 per cent. superior in economy to the directly-connected marine turbine, and only about 10 per cent. better propeller efficiency can be counted on due to slower revolutions of propeller. These two gains appear to be somewhat in excess of the loss in the transmitter, hence there should be a net gain in economy of something like 5 per cent.

In the case of the Melville & Macalpine reduction gear the promised increase in economy is much greater, since that gear has an efficiency of about 98 per cent.

An examination of the construction of the apparatus, and the fact that a considerable amount of water is contained in the transmitter, would make it appear that the saving of weight claimed for large powers is extremely doubtful. It will not be unusual to expect that some saving can be expected with small powers, like the 500-H.P. installation tested, but it is not extraordinary to expect a transmitter capable of handling 10,000 or 15,000 S.H.P. (with the *increased number of stages* and increased length and diameter that will be required) to be a very heavy and somewhat unhandy apparatus. The amount of water required for a large transmitter is likely to be considerable, and its weight will have to be considered. There may also be some doubt about the reliability of such a device where high powers have to be transmitted. It is one thing to reverse 500 H.P. and quite another to reverse 10,000 H.P.

If the device is reliable in service, its more adaptable field would appear to be a reversing mechanism for gas or oil engines. There is a good field for gas or oil engines of 200 to 800 H.P., and there are now many troubles in getting a suitable reversing mechanism. The Föttinger transmitter would appear to be just the thing to apply to small marine craft in conjunction with gas or oil engines.—H. C. D.

The transmitter was placed immediately abaft the steam turbine, though it may also be fitted before the stern tube at the end of the shafting. The general arrangement of the engines in the steamer is shown in Fig. 1.

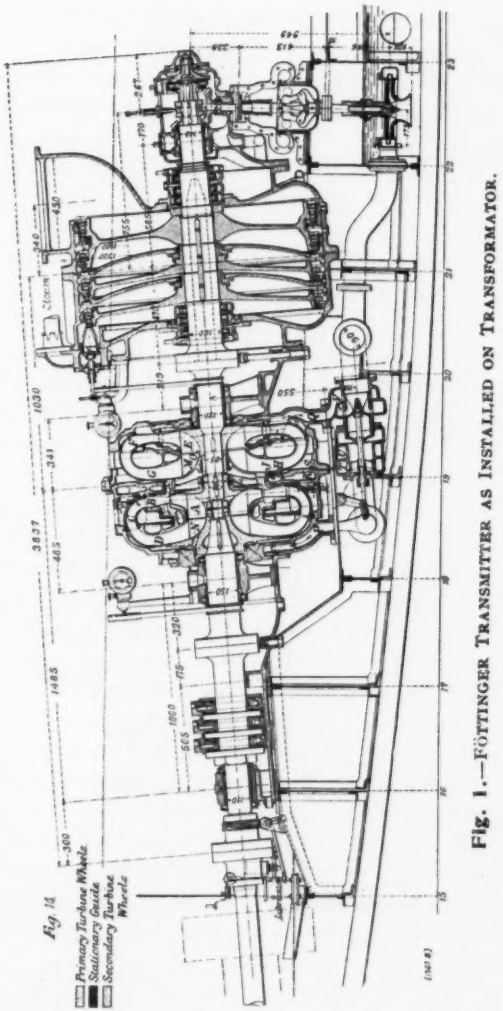


Fig. 1.—FÖTTINGER TRANSMITTER AS INSTALLED ON TRANSFORMATOR.

The Curtis turbine has four pressure stages, and each wheel has three rows of revolving buckets with two guide blades. The steam on the last wheel of the series impinges on the whole circumference. No diaphragm has been fitted between

the third and fourth stage, the idea being to get an axial steam thrust which is opposite to and balances the water thrust on the primary water turbine of the transmitter. The stuffing boxes, by the way, have been provided with carbon packing, which has been found very satisfactory, being soft and non-porous, so that they do not wear. The carbon blocks are in cast-iron casings. The bearings are of the ordinary turbine type, with forced lubrication, but without water jackets. There is a main governor on the same spindle as the auxiliary pump, to be referred to later, which prevents the turbine from exceeding the predetermined revolutions, in this case 1,900. The safety governor consists of an eccentric ring which comes into contact with a needle, throwing it upwards; in its travel it closes a throttling valve in the main steam pipe, not shown in our section.

The transmitter (Fig. 2) consists of two chambers of cast iron, the after one for driving the propeller shaft ahead at a reduced speed, and the forward one for driving it astern, also at a reduced speed. This arrangement may be altered so far as the disposition of the respective transmitters is concerned. Each of the transmitters is alike in principle. They consist of a primary water-turbine wheel mounted on the primary or steam-turbine shaft, and one or more—in this case two—secondary water turbines mounted on a secondary or propeller shaft, with one or more stationary guide wheels interposed at any place of the circuit. The wheels and blades are of bronze, and the wheels are keyed to the shaft as shown. The series of primary and secondary wheels and guide blades constitute a complete cycle, and the water moves in a constant compact flow, the primary wheels taking the water again immediately from the last secondary wheel; but guide blades may be interposed should this be considered necessary. The water, of course, impinges on the whole circumference of the turbine wheels. The water is given pressure and velocity in the primary wheel, which is coupled to the steam-turbine shaft, and delivers the water, in this case of ahead circuit, directly to the first secondary wheel; in this wheel only part of the energy,

especially velocity, is absorbed in driving the propeller shaft. The water then flows through stationary guide blades, which are connected to the casing, and, leaving these blades, passes through the second secondary turbine, which absorbs the re-

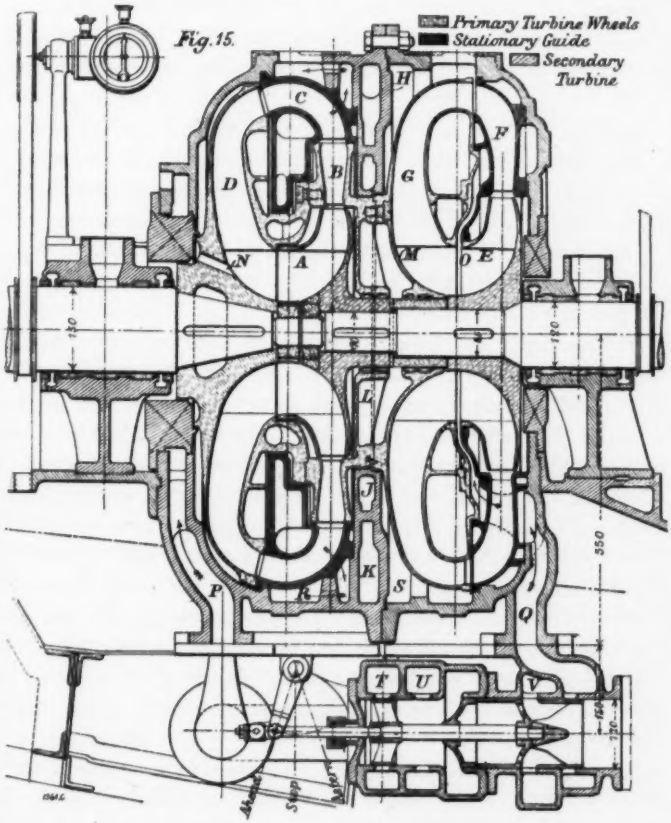


Fig. 2.—Section of the Föttinger Transmitter.

mainder of the energy; the water still flows at a certain velocity to the primary wheel. The first, and certainly the second, secondary wheel is reactionary. The blades of the primary water turbine are curved back, while those in the first second-

ary water turbine correspond to ordinary impulse water-wheel blades. The guide blades are similar to those in reaction water and steam turbines. The blades in the second secondary wheels are nearly radial and in some cases quite radial. The clearances are similar to those in ordinary steam-turbine practice. All spiral casings and connecting tubes to the primary centrifugal pump and secondary turbines are wholly dispensed with.

Taking first the ahead transmitter, shown to the left of Fig. 2, the part marked A is the primary water wheel, B the first secondary wheel, C the stationary guide blades, and D the second secondary wheel. B, it will be noted, is connected to D, which latter is mounted on the propeller shaft. In the go-astern transmitter—that to the right of the section, Fig. 2—the primary water wheel is marked E, the guide blades attached to the casing F. These, it will be understood, reverse the direction of flow of water to the secondary wheel G, which drives the propeller shaft astern, being coupled to the wheels B and D in the ahead transmitter, and through them to the propeller shaft. From G the water passes again to the primary wheel E. Between the ahead and the astern water circuit there is a diaphragm plate marked H, provided with two chambers J and K, to pass the water which would otherwise leak to the inoperative transmitter. These chambers communicate with drains leading to a small tank at the bottom of the boat for the supply of a small auxiliary centrifugal pump (Fig. 2). The amount leaking away was measured and was found to be about 1 per cent. of the mass of water circulating through the turbine wheels.

The medium for driving the transmitter, as we have already said, is fresh water, which is satisfactory from the point of view of friction and of density. No difference in efficiency has been noticed, even when large quantities of oil, saline, or other impurities have found their way into the water. On the runs from Stettin to Hamburg sea water was used. The transmitter is supplied with water by the small centrifugal pump driven from the steam turbine through bevel gear upon

an extension of the turbine placed at the forward end, as shown in Fig. 1. This absorbs about 0.5 per cent. of the power of the steam turbine.

The pump maintains a constant, but slight, pressure in the pipes between the supply tank and the maneuvering valve, which consists of several chambers within a casing. The pump serves for supplying the water when maneuvering, and for replacing the leakage. It is beneath the water level in the feed tank, so that there is always a suitable column to ensure efficient suction. Normally there is always sufficient pressure of water in the valve chambers to enable the water supply to be withdrawn from one or other of the two transmitters, and to pass it to the other. The water is fed into the ahead turbine through the port N in the secondary wheel D, this port leading it to the suction side of the primary wheel A for ahead action, while for astern action the supply is through some hollow guide blades F to the port O, adjoining the suction point of the primary wheel E.

In describing maneuvering it may be best to take first the reversing, which is accomplished by an admission valve controlling the flow of water to the admission nozzles or ports in the ahead or astern wheels. This valve is located in the chamber which supplies the water for making up leakage, and is under the level of the transmitter. The valve is horizontal, of the balanced-piston type, and the water either flows through the pipe P to the nozzle of port N for going ahead, or through the port Q, leading to the port for the astern water. The valves are mounted on one spindle, so that the port leading to P or Q cannot be opened unless the other is closed. Moreover, the seatings of the valves are so arranged that when P or Q is opened to pressure water from the pump, the other is opened to the exhaust pipe to the small reserve tank. The same valve is fitted with traps so as to pass off any water from one or both transmitters; thus the action of the valve permits the water to pass from one of these drain traps into the main drain system. The drain trap for the ahead circuit is marked R, and for the astern circuit S. These communicate with the valve chambers

T and U respectively. V is for filling the ahead turbine. Stuffing boxes are fitted forward and aft of the transmitter with metallic packings. These stuffing boxes are fitted on the hubs of the bearings in order to reduce the fore-and-aft space occupied. It will be noticed from Fig. 1 that a thrust bearing has been fitted.

Further experience of the working of this type of transmitter has resulted in improvements in the direction of simplification, although, so far as the action is concerned, no trouble whatever has arisen. The inventor has already designed an important modification whereby the ahead and astern transmitters are combined in one, so far as the primary and secondary water-turbine wheels are concerned, and both are always filled with water. The direction of rotation is altered by means of a sliding valve on a sleeve between the primary and the first secondary turbine, whereby different sets of guide blades are brought into operation for ahead and astern work respectively. This idea, which has also been patented by the Vulcan Company, offers considerable promise, and its practical application will be watched with much interest.

Meanwhile, however, it is interesting to note the results got with this installation. On the occasion of our visit to Stettin, to inspect the working of the "Föttinger Transformator," the trials were most satisfactory. The turbine was then running at 1,600 revolutions, the propeller shaft at about 270 revolutions ahead, the lever for controlling the valve for filling the astern water turbine and emptying the ahead water turbine was thrown over, and in 19 seconds the shaft was running at 250 revolutions astern. The return to full speed ahead was accomplished in even less time.

As to the transmission efficiency, Fig. 3 shows the efficiency curve at progressive powers. It will be seen from this that the efficiency of transmission in the ratio of 4.5 to 1 rises very rapidly, and from 600 revolutions of the primary shaft the efficiency is 78 per cent., advancing steadily until 1,250 revolutions is reached, when the efficiency is 83 per cent., at which it remains constant. Fig. 4 shows the results of the brake test

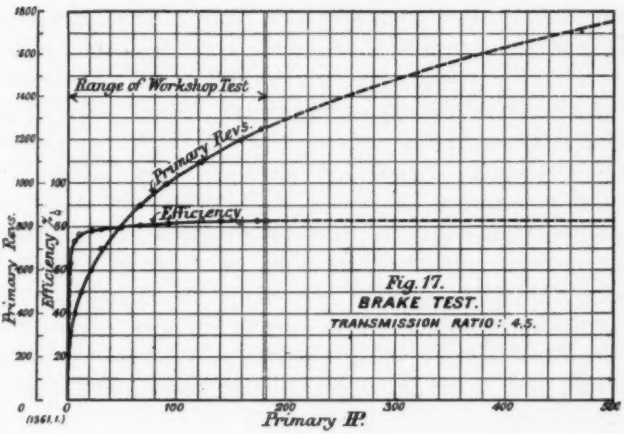


Fig. 3.

at 1,100 revolutions per minute. These were kept constant by the electric motor, and the secondary shaft revolutions were varied by the road on the Prony brake. The curve of the primary horsepower absorbed by the transmitter shows nearly

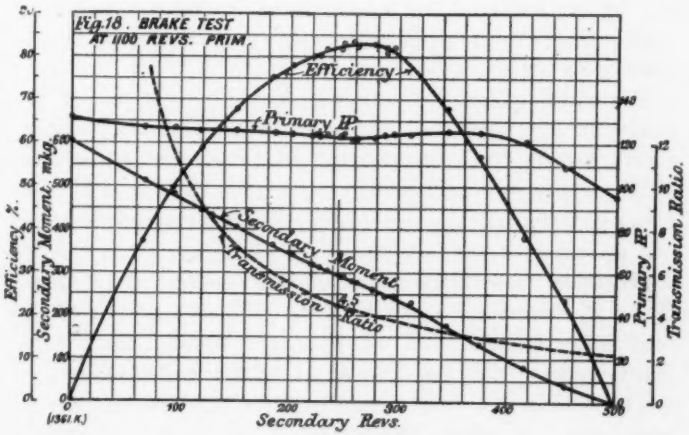


Fig. 4.

constant power rate for a very wide range of secondary speed. At the secondary speed corresponding to the designed transmission ratio, 4.5 : 1, *i.e.*, at about 250 revolutions, it seems to be a minimum, the efficiency a maximum, for this case of 122 primary horsepower = 83 per cent. Between transmission ratios of 5 and 3.7 it exceeds 80 per cent., between 6 and 3.5 it exceeds 75 per cent. This has been reached without any regulating devices. In case regulating devices, as in modern water turbines, should be employed, the range of highest efficiency will be still greater. This is of great importance, for instance, for regulating the transmission ratio at reduced speed of steam turbines at cruising speeds; a great number of patents for such devices and arrangements have been taken out by the Vulcan Company, and will be tried before long.

The secondary moment increases very rapidly when the secondary speed diminishes, and is more than twice the designed moment when the secondary shaft is stopped. This is very important for maneuvering. The diagram shows further that for a certain secondary speed—500 revolutions in this case—the secondary moment is zero, so that the propeller may race or the propeller shaft might break down without any danger to the parts of the transmitter, because the secondary wheels then would only attain about twice the designed revolutions, and never exceed this.

The efficiency may be further increased by utilizing the heat generated by friction in the water turbines to raise the temperature of the boiler feed water, and it is estimated that about 15 per cent. of the energy supplied to the transmitter is transformed by the friction of the water and other losses into heat, and the using of the boiler feed for filling the water turbines would raise the temperature by about 20 degrees to 25 degrees C. = 36 degrees to 45 degrees F., without special coal, the feed water being drawn off from the stuffing boxes in an easily-regulated way, and passed to the hot feed tank.

The system, it will be recognized, has very considerable potentialities. It enables the turbine to be overloaded without increasing the revolutions of the turbine, and secures for a

slow-running propeller all the advantages of the turbine, with its high efficiency and its freedom from oil in the condensed water, a very important factor in marine installations. The introduction of the reversing system permits of the continuous use of the economical high-speed ahead turbine, irrespective of ahead or astern steaming; and this, in ships which are frequently reversed, is an important factor. Moreover, the same power is available for going astern as for going ahead, while the adoption of large-diameter and large-area propellers gives greater propulsive power in reversing and maneuvering. In the event of any fracture of the shaft or the propeller involving over-racing, the transmitter acts itself as a relief governor to the turbine.

IMPROVEMENTS IN NAVAL ENGINEERING—PRESENT DEVELOPMENT AND FUTURE POSSIBILITIES.

By Lieutenant H. C. Dinger, U. S. Navy.

The feasibility of the application of new developments in naval engineering must be viewed from a military point of view. In this naval engineering differs from commercial engineering. Any development in naval engineering contrary to sound military principles is not of advantage and is ill-advised, as far as the attainment of naval efficiency is concerned. Hence matters which are of advantage commercially may not be of advantage in naval application, and every successful naval engineering proposition must be ultimately judged by the military advantage or disadvantage it produces.

How by change in design and improvement in methods of operation can the machinery of a naval vessel be made more efficient, and what, in the new and suggested applications to our machinery, is worth while? This is a field for our naval engineering personnel to exploit.

The main points of improvement lie in the direction of *economy of fuel, increase in efficiency and reliability, reduction in weight per power developed, cost and running expenses, and attending personnel.* All these are desirable, but sometimes one interferes with the other. Then the relative value is to be considered. Any operation or installation that is designed to secure economy whose cost is likely to measure up close to the value of the economy it will secure, must be left alone, since the habit of spending a dollar to save ninety cents never increased any one's bank account. There are also many things that may secure economy which are impractical for naval service, under the actual or probable

condition and with the class of attendants likely to be encountered. Thus, in considering the relative value of any device it is necessary always to consider any change in attending conditions that it may bring about, and the proposition must be viewed thoroughly from all sides.

The question of money cost must be weighed with the matter of military cost or gain. The money cost is a question to be considered chiefly on the ground of *amount of war or fighting efficiency per money expended*, and to this end is to be considered primarily in the cost of production, operation and maintenance. It must be fully considered in all cases where the application of a reduction in cost does not detract from military efficiency.

Reduction in weight and in consumption of fuel, oil and supplies has a military value as well as a money economy, since this affects not only the amount of money that has to be expended to provide a certain naval force, but also enlarges the capacity of this force for the purposes it is provided for.

A reduction in attending personnel is likewise a great military advantage, since this enables the same number of men to man more vessels and reduces the space and weight necessary for their accommodation, which may be used to more fully equip and provide for other features of the vessel.

Taking up the main propelling machinery, we have the proposition of using reciprocating engines, different turbines, or a combination of reciprocating engines and turbines. Propulsion of large men-of-war by gas engines has no immediate promise of successful application, due chiefly to the fact that a successful marine producer for work on bituminous coal has not yet been developed. Various means of electric drive, including the use of high-speed turbines and induction motors on the shafts, have also been proposed, but these systems are, as yet, not on a very practical basis and will, no doubt, need considerable experimental development to apply them in a practical manner to the field which is most promising for their use, that of the slow-speed merchantmen.

One of the great practical troubles that appear to manifest

themselves in connection with any arrangement of electric drive is the inability of making small and definite variations of speed; though the motors may be arranged for several speeds, say five or six, this will not do at all for *naval maneuvering purposes.* You must be capable of getting any desired number of revolutions exactly and directly and without unnecessary complication. Again, it is doubtful whether the double operation of generating electric power and then converting this again into mechanical power can be done as economically on a vessel as by the directly connected unit. Though the efficiency of both the generator and motor may be very high, it is doubtful whether more than 90 per cent. in each conversion can be counted on for continuous running in actual service. Hence, to come to the economy of the direct drive* the increase in economy of the turbine or gas engine and the increased efficiency of the propeller must be equal to a 20-per cent. saving to put the electric drive on an equal basis with the direct drive.

It is extremely doubtful whether as much gain as this can be obtained with the apparatus of the present day. The electric drive may be successful in certain fields and for certain conditions, but the propulsion of a battleship and the

*The economy of the marine turbine at or near full-speed conditions for large units is 12 to 13 pounds of water per S.H.P. and a propeller efficiency of 50 to 60 per cent. (British battleships and armored cruisers; *North Dakota* slightly more, but with a little better propeller efficiency due to lower revolutions.) Carefully designed reciprocating engines using superheated steam will give 12.5 to 13.5 pounds of water per I.H.P., with a propeller efficiency of 60 to 70 per cent. (*Delaware, Michigan, South Carolina*). These results have been obtained by actual installations, and, as far as the best performances of similar ships are concerned, about the same consumption of water per knot run on similar weights and space is obtained from turbines as with reciprocating engines at full speed, and with a superiority in economy for reciprocating engines at lower speeds.

Hence, a water consumption of 20 to 22 pounds per effective horsepower may be considered as a result which may be safely expected and probably bettered by the present directly-driven piston engines or turbines.

Now, if with the electric drive it is possible to obtain on board ship a kw. on 15.5 pounds of water per hour, and allowing an efficiency of 90 per cent. for the motor, one S.H.P. would be delivered for 12.8 pounds of water; and if a propeller efficiency of about 65 per cent. can be realized we have 19.7 pounds per effective horsepower, approximately the same as the turbine or reciprocating engine is now accomplishing. It may also be somewhat doubtful whether such good economy as 15.5 pounds of water per kw. can be secured with the turbo-generators when operating under the limitations as installed on board ship. The electric drive, therefore, promises to give us approximately the same economy at high speed that the present turbine or reciprocating engine gives. It is extremely doubtful whether the combination electric drive will give any saving in weight over present installations on large battleships, and it will require fully as much floor space.

requirements demanded—little weight, simplicity, extreme reliability, possibility of repair by vessel's own force—would seem to make the electric drive a matter of no great promise for securing increased advantages. It may be also accepted that it is not very likely that this will be used for naval work until after the system has been successfully applied in the commercial field where, on the moderate-speed freighter, it has a better promise of successful development.

It is believed that the gain by having a low revolution for propeller and a high revolution for the turbine can probably be more economically effected by means of reduction gear, several of which have been tried with promising results.

At present, for the various classes of naval vessels the following types of machinery seem to be those which offer the chief advantage for the conditions to be met:

For Battleships.—The combination of compound or triple-expansion reciprocating engines with L.P. turbine on four shafts, the reciprocating engines being placed on outboard shafts, turbines on inner shafts,—or improved turbines.

For Protected Cruisers (less than 22 knots speed).—The combination of reciprocating engines and turbines on three shafts.

For Scout Cruisers.—Turbines, on two, three and four shafts. For Parsons turbines, three shafts, with cruising turbine on center shaft only, would appear to give greatest simplicity and least weight. For Curtis or Zoelley turbines two shafts only would be employed, and the turbines fitted with cruising stages at H.P. end.

For Torpedo Vessels.—Turbines on three screws. Cruising turbine on center shaft only, or with Curtis or Zoelley turbines, one turbine on each of two shafts, cruising stages being fitted on H.P. end.

For Gunboats.—Reciprocating engines.

For large Colliers and Transports, Supply Vessels.—Combination of triple-expansion engines and L.P. turbines on three screws, as on S. S. *Otaki* and *Laurentic.*

Advantage of three shafts (where absolute isolation of one side from the other is not necessary).

Where this isolation is not considered essential, as on torpedo vessels, scouts and unprotected cruisers, three-shaft arrangement of turbines would appear to be the most advantageous, for the following reasons: The three can be stowed better; it saves in weight; better backing power, with less complication; there are fewer machines, hence less to take care of.

Cruising Turbines.—In the three-shaft arrangement (Parsons) it would seem most advisable to have but one cruising turbine at the forward end of the H.P. turbine. This would simplify the arrangement of piping to a great extent, would equalize the power on each side of the ship and would save weight.

With the ordinary Parsons arrangement of three shafts and a H.P. and I.P. cruising turbine on wing shafts, at cruising speeds the power is very unevenly distributed between the shafts. When using the five-turbine combination about half the total power is placed on one of the wing shafts. This unequal distribution must be overcome by the rudder, and good *propeller* efficiency under these conditions is not obtained. With the cruising turbine on center shaft the power is equally distributed on each side of the vessel, hence better conditions for effective propulsion are obtained.

With the use of impulse turbines of the Curtis or Zoelley type two shafts can be employed, and provisions for cruising stages made at the end of the main turbines. These turbines are simpler in their installation, but, as yet developed, appear to be somewhat heavier than the Parsons. Their economy, when well designed and taking into account the better propeller efficiencies, is about the same as the Parsons, while the danger of stripping blades appears to be less.

THE QUESTION OF THE COMBINATION SYSTEM.

The principal point where the turbine has the advantage in economy of steam per power delivered is that it works at a much lower actual back pressure, and the expansion of the steam can be more fully carried out. In the reciprocating

engine the actual back pressure at full power will on naval engines be from 4 to 8 pounds absolute. This back pressure can be reduced somewhat by more careful design of cylinders, valves and exhaust passages, but in any case it is doubtful if a practical back pressure below 4 pounds can be obtained on naval reciprocating engines. In the turbine a practical back pressure of about 1 pound and less is met with, and an expansion of 150 times. The amount of this saving may be better realized by an examination of the diagram, Fig. 1, the shaded portion showing the difference between a 4-pound back pressure and the ratio of expansion of 13 on the reciprocating engine, and the 1-pound back pressure with the turbine expanding down to 4 pounds. For the first portion of the operation steam in the turbine is not as efficient as it is in the cylinders of a reciprocating engine. However, the gain at the L.P. end is so great that it usually more than makes up for the deficiency at the first stages. This may also be shown by the fact that steam of same pressure as that entering the L.P. cylinder of triple-expansion piston engines will, in a L.P. turbine, do over half of the total work of the engine, while in the L.P. cylinder it only does one-third of the work. It may also be mentioned that the high efficiency of the turbine is only at or near full power, and at reduced power the economy drops very materially, and at very low powers, one-tenth full power and thereabouts, the economy is comparatively poor. With the marine reciprocating engine, on the contrary, the economy improves with a moderate reduction from full power, the most economical point being at about two-thirds power. With further decrease in power the economy drops slightly, but at about one-tenth full power the economy is about two times as good as with the turbine. By the use of cruising turbines the economy of the turbine system is improved, but even with these auxiliaries, at the low powers for speeds of 10 knots on a 20-knot battleship, the steam consumption of the turbine per power developed is over 50 per cent. more than that of the reciprocating engine.

The proposition of a combination system has its advantages

Fig. 1.

R. Part used by reciprocating engine.
T. Additional area used by L.P. Turbine.

based on the idea of utilizing the economy of the turbine for the last part of the expansion and the better economy of the reciprocating engine at the initial ranges, also that a reduc-

tion in power will not mean any appreciable reduction in economy till about half power or lower; for as the turbine, due to decreased power, becomes less economical the reciprocating engine becomes more economical. With a reduction below half power the economy is decreased, but not in as great a ratio as with the turbine or the reciprocating engine used separately. There is thus a continuous gain in steam economy effected by the combination. The installation can be arranged so that there is always a fair proportion of power developed in the turbine even at one-tenth full power, so that both steam efficiency and propeller efficiency are maintained all the way down.

Another very important point is that with the combination system a good degree of superheat can be used, and thus the gain in economy from this source can be realized.

Another very important gain over the turbine system is the gain in maneuvering power and propeller efficiency. Although the turbine is very efficient to develop power, owing to high speeds and small propellers employed, the efficiency of propellers operating at 400 to 500 revolutions per minute will be about 50 to 60 per cent., while one operating at 125 revolutions per minute will be about 60 to 70 per cent., so that there is a gain in propulsive efficiency of something like 8 to 10 per cent. for the combination over the turbine system. The combination system also will give a much better maneuvering power than does the turbine system, since large, slow-moving propellers are employed, and these are placed well outboard so that they give good leverage for turning. This is a most important advantage from a military point of view.

By the combination system at full power a propeller efficiency of about 58 per cent. can be obtained with a water consumption, using superheat, of about 10.5 to 11.5 pounds of water per S.H.P. With the turbine system the results would be about 52 per cent. propeller efficiency and steam consumption of 12 to 13 pounds per S.H.P. With reciprocating engines the results would be about 65 per cent. propeller efficiency with 13.5 pounds of water per H.P.

The combination is thus superior on the ground of propulsive economy to either the reciprocating engine or the turbine. Furthermore, it will weigh less than either the reciprocating engine or the turbine system. This decrease in weight over the turbine is approximately represented by the weight of the cruising turbines, and in the case of the reciprocating engine it represents the difference in weight of L.P. and a portion of the I.P. cylinders and their parts as compared with the L.P. turbine.

As compared with Curtis, Zoelley or other impulse turbines, the weight is also less.

The greatest saving to be accomplished is, however, in the boilers and fuel, for, due to the increase in economy, especially at low speeds, the boiler weights may be reduced about 10 per cent. and the fuel weights by about 30 to 40 per cent. to obtain the same cruising radius. This would mean, on a large battleship, a saving of 800 tons, or an increase in steaming radius of nearly 40 per cent. At present the reciprocating engine is more economical than the turbine at low speeds, but the combination is considerably more economical than either the turbine or the reciprocating engine. At very low speed the combination will give about the same economy as the best reciprocating engines.

The combination may be arranged in various ways, either on a three or four-shaft basis. If on a three-shaft basis, the powers should be divided so as to have about one-third on each shaft; if on a four-shaft basis, one-fourth on each shaft.

For the three-shaft arrangement the reciprocating engines would be triple-expansion engines having cylinder ratio $\frac{\text{L.P.}}{\text{H.P.}}$ of between 5 and 6, and designed so that steam to turbine at full power will be about 16 to 20 pounds absolute. Either a three or a four-cylinder engine could be employed, a four-cylinder engine being preferable on account of balance, although somewhat complicating the exhaust piping. This arrangement would be somewhat as in Fig. 2, being the arrangement employed on the *Otaki*.

The advantages of the three over the four-shaft lie in getting a large per cent. of backing power of the reciprocating engine and in a material gain in weight due to one large instead of two smaller turbines being employed. This system would be peculiarly advantageous on vessels where the engines are not required to be in separate compartments. Arrangements with the turbine on center shaft placed in its own compartment may be employed, but the disadvantage of having the operations in two compartments is present.

The four-shaft arrangement would require a reciprocating engine with cylinder ratios $\frac{\text{L.P.}}{\text{H.P.}}$ = between 2.5 and 5, and steam to turbine at full power about 24 to 30 pounds absolute. Either a triple-expansion engine or a compound engine could be used. The choice between a compound engine or a triple-expansion engine may rest on several matters. If the triple is used the ratio should be about 4; the reciprocating engine would do somewhat more than one-half the work. This would enable a relatively smaller turbine to be employed and would make the weight of the reciprocating engine more. If the compound engine is employed it would be advisable to use either three or four cylinders and a ratio $\frac{\text{H.P.}}{\text{L.P.}}$ of not greater than 3.

The difference between employing a triple and a compound would then result in having the larger part of the power on the turbine in case a compound engine were employed, and on the reciprocating engine in case the triple-expansion engine were used. Between the two there does not seem to be any very material change in economy, one way or the other, and the use of the compound engine would appear to have some advantage in the matter of weight; the use of smaller cylinders and reciprocating parts would be of some advantage as far as overhaul and attendance are concerned. On the other hand, the triple-expansion engine would give somewhat better power for backing. An arrangement of a 4-cylinder compound engine and L.P. turbine is shown in Fig. 3.

Fig. 1.

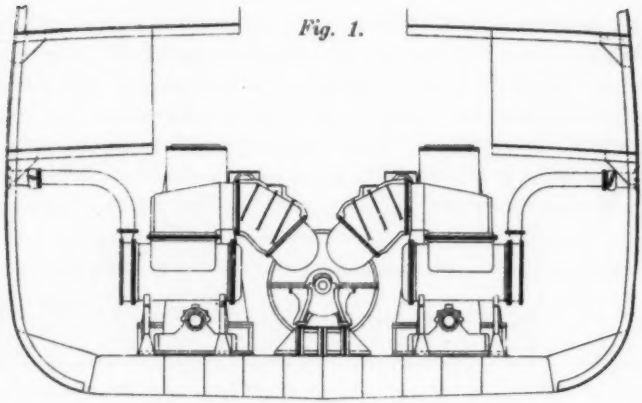


Fig. 3.

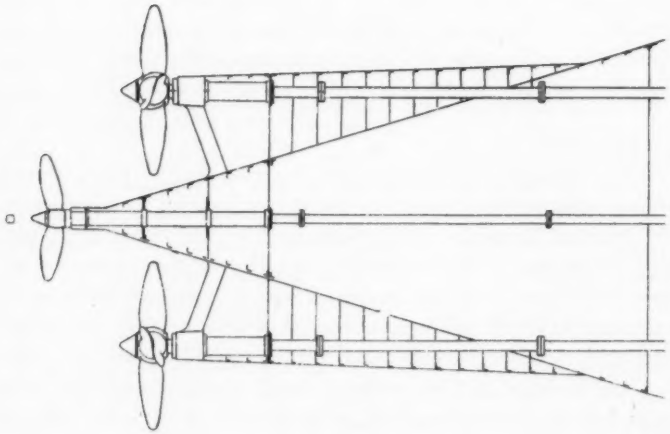
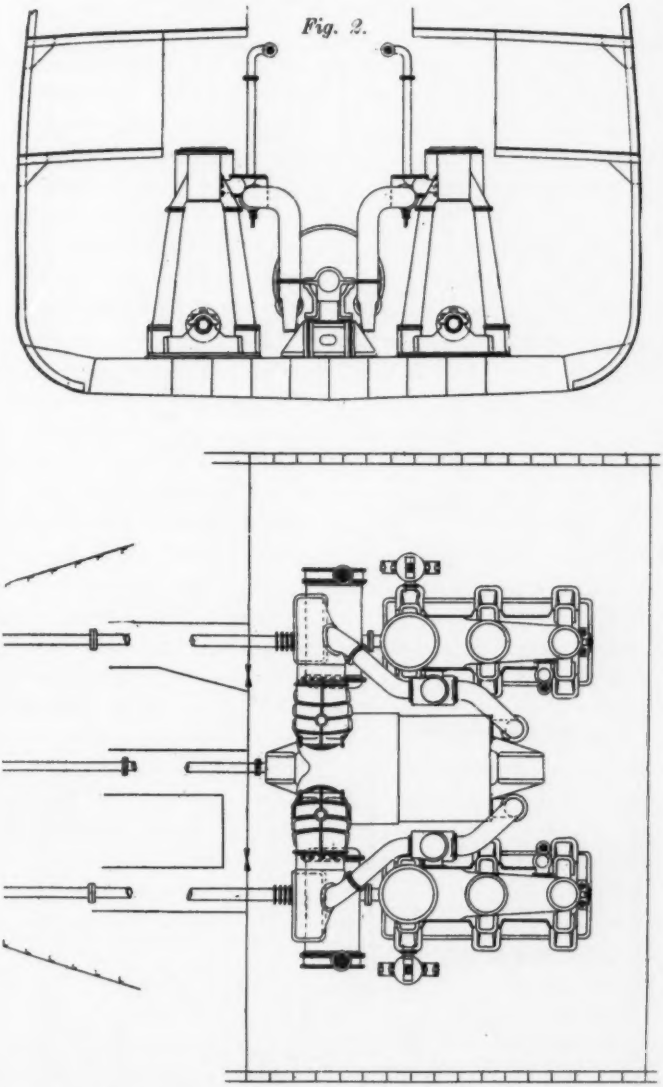


Fig. 2.—ARRANGEMENT OF COMBINATION

Fig. 2.



SYSTEM ON STEAMER "OTAKI."

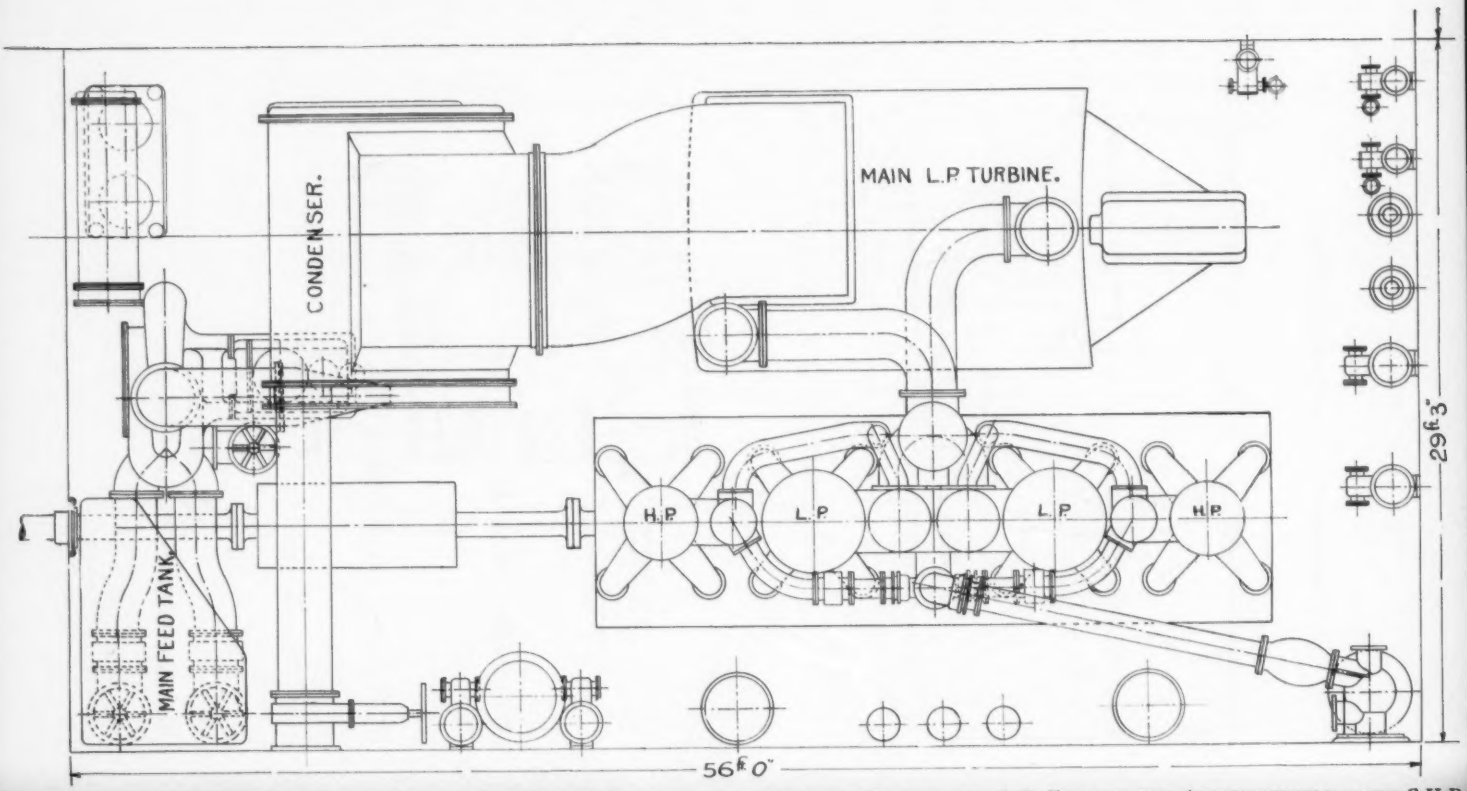


Fig. 3.—Plan Starboard Engine Room. Combination, Compound Reciprocating Engine and L.P. Turbine for Approximately 30,000 S.H.P.

For backing power with the combination-system four-shaft arrangement a backing turbine may or may not be employed. As the reciprocating engine can be arranged so that it will give one-half power, it would not appear necessary to make use of a backing turbine, and hence the weight and cost of the astern turbine can be saved. However, if it is considered desirable to have the added backing power, the backing turbine can be fitted.

For 30,000 H.P. the triple-expansion engine would be either a 38×55×85-inch or 38×55×two of 60-inch. The compound should be a 38×two of 55-inch or two of 28-inch inch×two of 55-inch, all with 48-inch stroke. The question of whether the relative advantage of superior balance of the four-cylinder engine would be worth the increased weight and space is a question meriting careful attention. By using one instead of two H.P. cylinders the clearance can be made lower, a point of great advantage.

With such a combination, using steam of 500 degrees F. and exhausting from reciprocating engine into the turbine at about 30 pounds absolute, the engine would give a steam consumption of about 21 pounds per S.H.P. The turbine using this exhaust and a vacuum 28.5 inches would give a steam consumption of about 23 pounds per S.H.P., or about 11 pounds of water per S.H.P. for the combination. Counting 60 per cent. propeller efficiency, this gives 18.3 pounds of water per effective horsepower. At three-fourths power the economy will be about the same, the piston engine being slightly more economical and the turbine slightly less.

The combination will produce approximately the following economical results at lower powers:

	Water per S.H.P.	Propeller efficiency.
One-half power,	12.0	61 per cent.
One-fourth power,	12.5	61 per cent.
One-sixth power,	13.5	62 per cent.
One-eighth power,	14.5	62 per cent.
One-tenth power,	16.0	62 per cent.

FIG. 3.—PLAN STARBOARD ENGINE ROOM. COMBINATION, COMPOUND RECIPROCATING ENGINE AND L.P. TURBINE FOR APPROXIMATELY 30,000 S.H.P.

These calculations are based on actual results of water used by L.P. marine turbines and by H.P. and I.P. cylinders of reciprocating engines in actual service, and from the performances of S. S. *Otaki* and *Laurentic*, which are smaller installations, using lower steam pressure without superheating, and, therefore, there is no reasonable doubt that the above performances can be realized and probably exceeded by a good margin.

In view of the superior economy, increased reliability and the saving of weight that is accomplished by the use of the combination system, and the superior maneuvering powers, there does not appear to be much question as to its superiority for moderate speeds over either the reciprocating engine, turbine arrangement or the electric drive. Its adoption means an increase of at least 40 per cent. in steaming radius at lower speeds over the turbine outfit and about 10 per cent. over the reciprocating engine alone.

The point of objection raised is that the combination retains many of the practical disadvantages of the reciprocating engine; but the reciprocating engine, properly designed and built and when fitted with forced lubrication, does not have very many practical disadvantages, and is remarkably free from breakdowns, and really much more reliable than turbines have yet proven themselves to be.

Turbines of the impulse type, which are capable of being arranged as independent units on each shaft, have an apparent advantage in simplicity and give satisfactory results for maneuvering, but as yet these turbines are both heavier and less economical than the combination system would be.

TURBINE REDUCTION GEAR.

Another new development, which now only needs the demonstration of its operating success and reliability in large units, is a reduction gear for marine turbines. The Melville and Macalpine type of reduction gear was described in the November, 1909, JOURNAL, and was recently tested in the

presence of representatives of the Bureau of Steam Engineering. Another hydraulic reduction gear (Föttinger's) is being tried out in Germany ("Engineering," Nov. 5, 1909), see page 83, of present number. The fitting of the reduction gear will enable much higher rotative speeds to be applied to the marine turbine while at the same time allowing propeller speeds which permit of a high degree of efficiency.

The higher rotative speeds will also allow us to secure a material gain in the economy of the turbine and lessen the weight. The success of these reduction gears will depend mainly on the reliability in service and whether appreciable wear can be avoided. The use of reduction gear need not, however, detract from the advantage of the combination system of using reciprocating engines in conjunction with turbines. The combination, in fact, makes the use of a reduction gear much simpler, since, if the combination is employed, the turbine and its reduction gear can be arranged for ahead motion only, and thus the weight and space made necessary by reversing arrangements is saved.

COMBINATION SYSTEM IN CONJUNCTION WITH REDUCTION GEAR.

It is not unlikely that reduction in cylinder clearance to about 10 per cent. and special refinement in the design of steam ports and passages, free from sharp turns, edges, irregular openings, etc., that a very material increase in economy can be effected, so that, with a reciprocating engine using steam at 500 degrees F. a water consumption of 10 to 11 pounds of water per I.H.P. may be realized. (12.7 was realized on 19-knot trial of the *Delaware*. The engines of the *Michigan* and *South Carolina* are no doubt more economical, coal consumption is considerably less, cylinders better proportioned, but no water-consumption data was taken, so there are no definite figures.) Of course, this means special care and attention in making the design, in building the patterns and in constructing the engine. This adds considerably to

the cost of building, and this is, perhaps, the principal reason that those matters often do not receive due consideration. However, when it is considered that the extra steam consumption (which the special care and small extra expense can avoid) is going on every day the engine is used, it is plain to see that the extra cost will be paid for in a very short time. By having a compound reciprocating engine taking steam superheated to 500 degrees F. and with clearances reduced to 8 or 10 per cent., with well-designed ports and valve gear an economy of 20.5 pounds of water per S.H.P. can be obtained, with exhaust at 30 pounds absolute. With a high-speed L.P. turbine fitted with reduction gear taking this exhaust, an economy of 18.5 pounds per S.H.P. can be obtained. So that the final result will be a S.H.P. on 9.7 pounds of water. This represents the possibilities, and if the installation is carefully designed and constructed there is not much doubt but what this result may be closely approached in practice.

FORCED LUBRICATION.

This is one of the most important improvements that can be made to the reciprocating engine, and the application of this system to naval engines makes a big advantage gained for the reciprocating engine in its competition with the turbine. With forced lubrication the reciprocating engine may be made practically as reliable with regard to lubrication, and almost as economical in use of oil, as the turbine. This system is also being used on large British merchant vessels. This important subject is fully treated and described in the February, 1909, number of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.

No doubt with forced lubrication it will be possible to increase rotating speed of journals and make reductions in heavy surfaces that would be dangerous with ordinary lubricating systems. (See page 58 of present number.) It has also been found in actual practice that by the use of forced lubri-

cation a very considerable increase in the number of revolutions of the engine was secured, thus showing that the frictional losses ordinarily met with were greatly reduced.

One of the particular advantages of the forced lubrication is the saving of oil. This means much to a man-of-war, as the radius of oil supply is to be considered as well as that due to the fuel supply. The frictional losses on a large marine reciprocating engine are usually from 4 to 8 per cent. of power developed. If forced lubrication reduced this friction by one-half it means a gain of 2 to 4 per cent. in shaft power.

IMPROVEMENTS IN RECIPROCATING ENGINES.

Cylinder Proportions.—For good economy at high power and using high-pressure steam (250 to 300 pounds), a large ratio $\frac{\text{H.P.}}{\text{L.P.}}$ must be used. This ratio is necessary in order to secure the proper degree of expansion and in order that the terminal pressure in L.P. cylinder may be as low as possible. For the above pressures the cylinder volumes should be about as 1–2.5–10, and the L.P. cylinder should be proportioned for a referred mean effective pressure of about 36 pounds.

Reduction in Clearance.—Nothing is more important in securing economy of the reciprocating engine than the reduction of clearance. Numerous engines have been built where the clearance of H.P. cylinder was more than 30 per cent. By considerable care and attention it has been reduced to 15 per cent. for H.P., 12 for I.P. and 10 for L.P. This has been brought about by making the clearance passages of uniform cross section, cutting out all corners and pockets, moving valves close to cylinder and using long valves, so that the valve ports come abreast the cylinder ends.

Reduction of friction in passages from valves to cylinders should also be secured. All sharp and irregular cross sections and square corners are to be avoided. All the passages should have rounded corners and care should be taken that uniform cross-sectional area is secured. A condition is often

met with where the cross-sectional area at certain points is much larger than at others. This is a clear case of unnecessary clearance. If the passages are so formed as to give a smooth flow they can be reduced to a very large extent. The experiments of Guthermuth, republished in May, 1904, number of the JOURNAL, indicate that the nature of the openings and orifices has a tremendous bearing on the capacity of steam flow.

The configuration of the clearance spaces, the ports and passages, are matters of greatest importance in securing improved economy, and when the results with recent improved engines are compared with engines previously built it is most surprising that careful attention to these matters should not have been more general during the last ten years.

Other matters tending toward economy are:

(*a*) A more efficient and thorough lagging. It is probable that thorough experiment with various new forms of cellular lagging will show that very considerable improvement per weight of lagging can be secured.

Very little definite data on this matter seems to be public, nor does it appear that much material progress in this field has been made in the last decade.

(*b*) The use of slide valves instead of piston valves on L.P. cylinders. Slide valves, to begin with, allow less clearance, and they are more apt to be tight than are piston valves.

(*c*) Efficient balancing devices for valve gear in reducing the inertia tend to secure smoother running and help economy.

BOILERS.

Turning to the boilers, the questions are: Mechanical stokers, coal or oil fuel, superheating, air heating, and improvements in methods for maintaining boilers in efficient condition with minimum amount of repairs.

The steam pressures for both naval and merchant service is now between 200 and 300 pounds above the atmosphere, and it is not likely that there will be any considerable variation in the near future from the present pressures used.

Mechanical stokers have been used to a considerable extent on the Great Lakes, but the mechanical stoker for marine work cannot as yet be considered a commercial success. For naval work their installation meets with even greater disadvantages. To begin with, the extra weight and space needed for the installation of mechanical stokers is a difficulty hard to surmount. It will also be a problem to adjust the stokers to the high rates of combustion and the great and sudden variations in power that are at times required for naval service. About the most promising mechanical-stoking apparatus for naval work is that appearing in the Notes of this number. On the other hand, oil fuel gives all the advantages of mechanical stokers and requires no increase in weight, but makes a considerable saving in the weight of necessary fuel. (The objection to oil fuel is the unreliable supply and high cost.)

The greatest saving in weight, while still maintaining the ability to steam with coal alone when necessary, is accomplished by making a portion of the boilers for oil fuel only and another portion for coal only. This will make it possible to use either coal or oil as they are available, and at the same time lessen the boiler weights considerably below that of an installation where either coal or oil can be used on all boilers. By doing this better combustion of oil is secured, since we are not yet able to burn oil in combination with coal and secure thoroughly satisfactory combustion for the oil. These questions were extensively treated of in the article, "Oil Fuel for Naval Use," February, 1909, J. A. S. N. E.

The use of oil alone for sea-going cruising vessels, in the light of the present distribution of oil supply and the cost of this fuel, does not appear to be advisable. But the use of oil to some extent, on account of its very evident advantages, is to be urged.

SUPERHEATING.

This is, perhaps, the greatest source of economy introduced in marine practice since the adoption of the triple-expansion engine. There is now no question concerning the enormous

gain that can be accomplished, nor is there any difficulty attending the practical operation of superheaters. The gain due to the use of superheated steam over saturated steam, using a boiler pressure of 200 to 300 pounds, is from 10 to 12 per cent. for 100 degrees superheating, and about 6 per cent. for 50 degrees. The use of the superheater need not add to the boiler weights, since the increased economy bears in a greater proportion to the total steam consumption than does the weight of the superheater to the total boiler weights. Thus the total weight of the boiler installation can be slightly less than the boiler weights without superheaters for the same power.

The chief difficulty that may be encountered with the use of superheat is that, owing to the higher temperature used, the piping and fittings must be of material that will stand the higher temperature. Therefore, it is advisable to use steel piping and steel or monel-metal valves and fittings. However, if the temperature of the steam does not exceed 500 degrees F., good, heavy composition or bronze fittings will give satisfaction, and may be used for the small fittings where it is difficult to employ steel castings. Another place where difficulty from superheating is feared is in the H.P. valves and cylinder. This difficulty is also one that can be overcome. It is a problem of having strong, properly designed and well proportioned piston and valve packings and proper lubrication of the wearing surfaces. The lubrication of the surfaces can be accomplished by a graphite feeder and the frequent coating of the wearing surfaces with graphite grease when opportunity offers between runs.

These difficulties, of course, become greater as a higher temperature is used, but experience has demonstrated that a temperature of 500 degrees F. may be dealt with in a thoroughly satisfactory manner and without the need of special precautions or the installation of extra safeguards.

The type of superheater should be one embodied in the boiler, with as few extra fittings as possible, and the superheating surface should be placed in the middle portion of the travel of the gases through the boilers. If placed so that the

gases strike the superheating surface on leaving the furnace there is danger of burning out the superheater, while if they are placed in the uptakes the gases are not sufficiently hot to give a good degree of superheat. In connection with this, there is the relative advantage of the bare-tube or protected-tube superheater. The B. & W. superheater as fitted on our

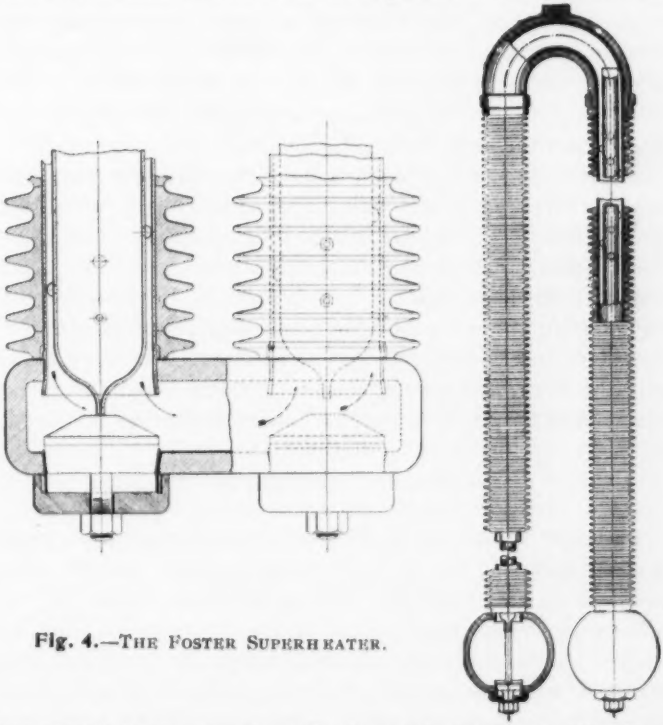


Fig. 4.—The Foster Superheater.

late vessels is an example of the bare-tube heater.* The Foster superheater is an example of the protected type. The Foster heater, by means of the cast-iron rings, gives a larger heating surface for the gases to act on, and the rings form a reservoir of heat which will conduce to securing a more uni-

* This heater and its details is fully illustrated in the November, 1909, number, pp. 1295–1297, "Test of superheated steam on yacht *Idalia*."

form degree of superheat. The inner tube of the Foster heater causes the film of steam to fill only the annular space, thus making the surface more efficient. The cast-iron coverings also prevent any possibility of superheating tubes being burned.

For the same steam surface the Foster heater is, of course, considerably heavier, and choosing between the two, it is a question of whether the increased efficiency of the surface due to the use of inner tube and the value of a uniform temperature are sufficient to make up for the disadvantage of the extra weight. A comparative experiment test of the two types under the same conditions would be very interesting.

Separately-fired superheaters have the objection that they require more floor space and a special operating force, and, for the same capacity, are likely to weigh more than when made a part of the boiler; also, the discharge gases for the separately-fired superheater are likely to be considerably higher than those for a boiler fitted with superheater. Unless some special arrangement of economizer is fitted there will, therefore, be a greater loss for the separately-fired superheater than for one contained in the boiler casing.

AIR-HEATING ARRANGEMENTS.

The chief drawback to air-heating arrangements on men-of-war appears to be the extra weight needed and the relatively small gain possible. The possible gain is also largely influenced by the temperature of the gases on leaving the boiler. If the boiler is not well baffled the temperature of the gases will be high, and this will enable more air heating to be done. If, on the other hand, the baffling is efficient and there is a low uptake temperature, say 450 to 500 degrees F., the amount of air heating will not be very great. Hence the greatest gain for air-heating installation will be found on types of boilers where the gases have a high uptake temperature.

However, it is believed that such air-heating arrangements

as will not add appreciably to the weight or complication may be used with advantage. It is the practice, to a great extent, to have air spaces at the sides or tops of water-tube boilers, these air spaces being designed to serve as portions of the boiler lagging. Now, if these spaces are arranged so that the air being taken in from the fireroom is gradually heated and finally discharged to the furnace, either through or below the grate, as may be most desirable, the heat that this air absorbs from the boiler will be saved, its effect as lagging will be increased, and practically no weight and very little extra gear will be added.

This is especially desirable in the case of express boilers, and numerous types utilize this feature to some extent. It is also possible to add this feature to the uptakes in the boiler room and thereby obtain a very considerable air-heating surface. It appears that the tops and back and portions of the side of the B. & W. boilers could be fitted with an air casing which would tend to prevent the heating of the firerooms and add somewhat to the economy. The gain in economy would, perhaps, not be more than one or two per cent., but this even would be well worth while, since the extra weight would be saved in a few days' steaming, and the extra cost would be almost negligible; besides this, the need of extra lagging on uptakes and the special air casings under the protective deck might be obviated.

BOILER PRESERVATION.

One of the chief fields for naval boiler improvement is in insuring their durability and freedom from repairs, which means mainly a reduction in boiler corrosion. There is no great danger from salt, since tight condensers are now the rule, but there is still the danger from corrosion due to air in the boiler and the burning of tubes due to deposits of oil and dirt.

The danger from acid corrosion may be overcome by keeping the water alkaline. It appears that there are other sources of acid than that due to the salt water. It has been

found that freshly-distilled water under certain conditions is acid. For this reason, then, it is necessary not only to keep the boiler water alkaline, but the water in the feed tank should be even more carefully watched, since, though a sample of the water for a large boiler may be alkaline, in certain portions of the boiler the water may be acid. The best method of guarding against any chance of acidity is to have a regular and continuous injection of alkali into the water being delivered to the boiler; in other words, to effectively prevent any acid water being sent into the boiler.

Another source of boiler troubles is oil and dirt deposit. This deposit reduces the efficiency of heating surface and may cause tubes to become overheated and bulge or collapse. The remedies to be applied are, first:

Prohibition of oil for internal lubrication;

Use of filters and grease extractors;

Use of alkali or boiler compounds for cutting the oil and keeping it in solution and enabling it to be blown out through surface and bottom blow. Sal soda, caustic soda, kerosene, turpentine and various special boiler compounds are used for this purpose. These are more or less effective, but are probably crude means as compared with special compounds prepared after a study of the physical and chemical action required for certain given conditions. Boiler compounds are largely used on shore installations where the water contains special deposits or where oil is present in feed water. The use of special compounds has, however, not been very extensive in naval work, but it does seem that some experimenting in this field would develop compounds by means of which the necessity for very frequent and costly cleaning of boilers would be avoided to a great extent.

AIR CORROSION.

It is not unlikely that a very large part of the boiler corrosion found in our naval boilers is due to air. For this reason every possible precaution should be taken for (1) keeping air

out of the feed discharge, (2) causing it to be liberated in the steam and space and not in the tubes of the boiler.

In order to get rid of air in feed water it has been proposed to install a large air vessel on feed heater, and by means of an air cock to blow the air out as it collects; also, any connections by which air is introduced into the feed suction must be avoided, and leaks in feed suction prevented.

Air contained in the feed water can be caused to be liberated in the steam space by having the feed discharge in a fountain spray above the water line instead of through an internal pipe below water level. As air collects just above the water line in the boiler it would seem that the regular use of surface flow to drive it off would also be advisable.

FEED HEATERS.

Possible improvements in this apparatus seem to lie in the use of spiral or corrugated tubes and the use of what is known as the film heater, for the purpose of securing more efficient heating surface, as this necessitates less surface and weight. Improvement in method of operation lies in use of additional heaters which will enable a temperature higher than that obtainable from the auxiliary exhaust to be obtained.

Spiral or corrugated tubes are unquestionably more efficient than plain tubes, and the film principle, together with the use of corrugated surfaces, enables very material reduction in surface to be made. The Reilly heater uses spiral, while the Shutte-Koerting corrugated film heater and the Wainwright heater use corrugated tubes, all three being used in late naval work.

It is generally found that at cruising speeds on naval vessels there is more auxiliary exhaust than is required to heat the feed water to 210 degrees. Now, if a portion of the auxiliaries exhausted at a high pressure, say 40 pounds absolute, and were led to a heater into which the feed was delivered after leaving the ordinary exhaust-steam heater, operated at about 20 pounds absolute, a higher feed temperature could be

obtained. If the exhaust of feed pumps and circulating pumps at 40 pounds absolute were led to an intermediate heater a temperature of approximately 250 degrees F. might be obtained instead of the 210 degrees now secured. This would improve the economy something like 2 or 3 per cent. It would, however, entail some additional weight, and the added complication in exhaust piping and feed connections offers considerable objection.

Another arrangement would be the use of a series heater in two parts—one to use the auxiliary exhaust at about 20 pounds absolute, giving a feed temperature of 210 degrees F., and a high-pressure heater, fed with steam from the first receiver of the engine, the pressure being approximately 100 pounds absolute. With 10 per cent. of engine steam used for this H.P. heater, a temperature of about 350 degrees F. should be obtained, resulting in an increase in economy of 3 to 4 per cent. The increase in weight for this installation above the ordinary exhaust heater would not be more than a few tons on a battleship, a weight made up in a few days' steaming. There is, however, the increase in steam and feed connections.

If steam of boiler pressure is used instead of that from the first receiver, a still higher temperature of feed can be obtained, but more trouble with heater is likely to be encountered and will have to be considerably heavier.

Another and more feasibe proposition is to use heating coils in the delivery chamber of the exhaust steam heater; the coils to be supplied with steam from (a) evaporator coils, (b) I.P. receiver, (c) drain and from separator drains of steam pipes. Such coils may be arranged for in the discharge chamber of the heater, with very little additional space and very little extra weight or piping. Such an arrangement is indicated in sketch. The part A is designed like an ordinary single-pass closed heater using special corrugated tubes. The discharge chamber is made slightly deeper, and in it are placed a series of pipe coils. The first series will take their steam and condensed water from the evaporator-coil drain, the pressure of this being between 60 and 80

pounds absolute, with corresponding temperature of 293 to 312 degrees F. The second series will take their steam from the I.P. receiver, the pressure here varying from 50 to 115 pounds (depending on speed), and corresponding temperature

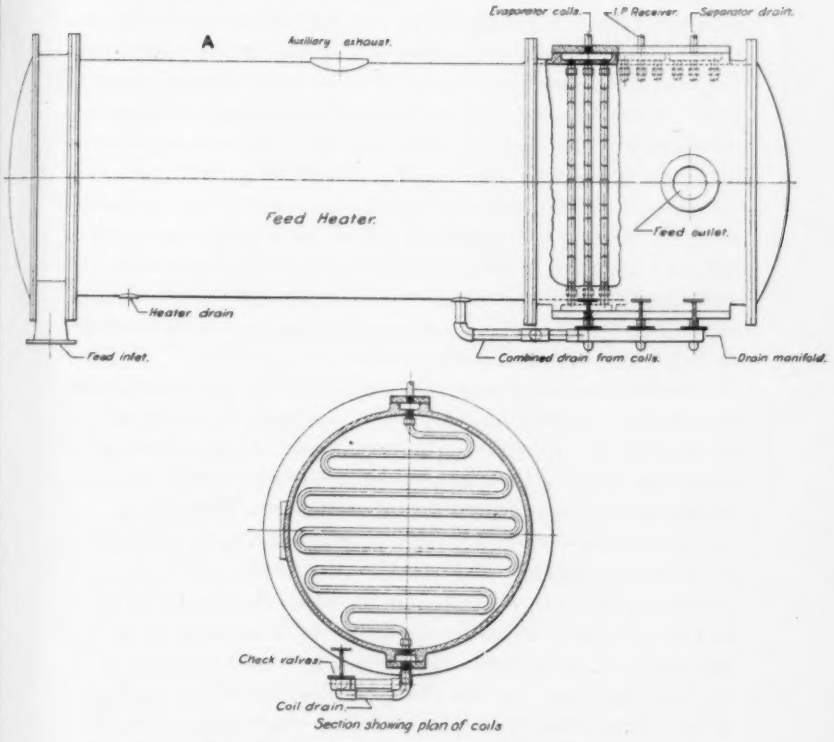


Fig. 5.—FEED HEATER FITTED WITH EXTRA HEATING COILS IN DISCHARGE CHAMBER.

280 to 338 degrees F. The third series will take steam from the steam-separator drains, the pressure being 200 to 300 pounds (depending on speed), and corresponding temperature 381 to 417 degrees F. By means of these coils, using 2 to 3 per cent. of the total steam from the evaporator-drain coils,

heating water approximately to 235 to 245 degrees F.; using 5 per cent. of total steam in I.P.-receiver coils, heating water to approximately 280 to 320 degrees F.; using 3 per cent. of total steam in separator-drain coils at boiler pressure, a temperature of 340 to 360 degrees F. for feed water might be secured. Since a considerable portion of the heat from the separator drains and evaporator drains is now lost, only about 6 per cent. of the total steam is taken from the engines and used in the heaters, and the expected increase in economy obtained by the use of the heaters in this manner would probably be about 5 per cent. This, considering the very slight increase in weight, and that there are no extra complications to speak of, and the beneficial effect on boilers, would seem to be an installation that it would be advisable to try out.

The same idea can be carried out with the Reilly heater, or most any other type of heater, by placing similar or spiral coils in the discharge water chest.

Large air vessels should be fitted on discharge chambers for the double object of reducing shock and jar on the heater and of serving as an air trap by means of which considerable quantities of air may be taken out of the feed water and thus prevented from having its corrosive effect on the boilers.

Operation of Heater.—Where the auxiliary exhaust is more than sufficient to maintain the feed water at the temperature due to the exhaust it is advisable to open the exhaust to the L.P. receiver to an extent such as will allow the surplus and exhaust to do work in the L.P. cylinder or turbine while still having the feed at the temperature that the pressure in the exhaust line will give. On the other hand, if there are arrangements for using the auxiliary exhaust for operating the evaporators the excess exhaust could be used there.

DISTILLING PLANT.

This service uses about one-third of the coal consumed on a naval vessel in port, and for this reason it is a field where improvement will count.

The most promising features for securing improvement are: (1) Operate distilling plant in double or triple effect; (2) Use vapor from the L.P. evaporator for heating evaporator feed; (3) Provide efficient separators and proper baffles in order to reduce losses due to priming on starting up; (4) Make arrangements whereby the coil drain after leaving evaporator coil may be caused to give up its heat to a useful purpose. This may be done by having this drain water used in the feed-water heater, as mentioned under Feed Heaters, or by having it heat the evaporator feed to a temperature corresponding closely to the temperature of water in evaporator shell.

One pound of steam in ordinary single effect will distil .8 pound of water; with double effect, about 1.5 pounds. If we heat the evaporator feed to 210 degrees F. by means of the vapor, and from 210 to 255 degrees F. by means of the evaporator-coil drain water, about 175 B.T.U. for each pound of water distilled will be saved. Hence, where before, with double effect, it required 730 B.T.U. to distil a pound of water, it will require only 730—175, or 555 B.T.U., a saving of about 24 per cent. by the use of the heaters for evaporator feed. This will give about 1.8 pounds of water per pound of steam, or about 15 pounds of water per pound of coal, or 1.8 gallons per pound of coal. Under these conditions it will require 5,882 pounds of coal, or 2.62 tons, to distil 10,000 gallons of water. This is about the best practical result, using evaporators in double effect, evaporator-feed heaters, and discharging vapor from L.P. coil to distillers or feed tanks and vapor from L.P. to distiller. The actual service results, allowing for extra radiation losses, leaks, time for blowing down, etc., will probably show a coal consumption about 30 per cent. above this. By the use of double effect and the employment of evaporator-feed heaters the coal for distilling plant can be reduced at least one-half from that used for single effect without evaporator-feed heaters. This represents conditions where vapor from the L.P. evaporator is distilled. This is necessary for drinking water. For make-

up feed while under way, the vapor from the L.P. can be turned into the auxiliary exhaust; and, with the auxiliary exhaust either turned on feed heater or on the L.P. receiver, still better economy is obtained, since the heat contained in the vapor is utilized.

It would then seem that the best economy would be to get the make-up feed as far as possible while steaming and do the distilling in port. Also, if any considerable steaming is done it would be advisable to use fresh water for cleaning, scrubbing, etc., from the reserve-feed tanks, for the reason that this water can be more economically produced than that which is gotten through the distillers.

Thus with the addition of a few simple devices, whose weight is almost negligible, the consumption of coal for distilling and make-up feed can be cut down to one-half, and the make-up, by using vapor on the auxiliary exhaust, with all excess on the L.P. receiver, can be still more economically produced.

The project of using the vapor of the evaporator in a feed heater through which the main feed delivers may also be used, and this would be most promising if the vessel steamed all the time and the auxiliary-exhaust steam were insufficient to heat the feed water up to the limit of the pressure carried in the auxiliary-exhaust pipe. However, the auxiliary exhaust on a naval vessel, except at high speeds where the F.D. blowers are electrically driven, is usually more than enough for feed heating. This system would produce economy principally at such times as the auxiliary exhaust was insufficient to keep up a temperature of 200 degrees. This condition does not obtain on most naval vessels at ordinary cruising speeds. It may be that 200 degrees feed is not obtained, but this condition is generally not due to insufficient auxiliary exhaust, but to defective operation or installation of feed heaters.

But as the auxiliaries are becoming more economical and as the blowers are electrically driven, there is on our latest battleships insufficient auxiliary exhaust at three-fourths power and above to enable a feed temperature approximating

to the temperature of the auxiliary exhaust to be obtained. Here the heat that could be added by passing the feed through an evaporator vapor-feed heater would add materially to economy at the higher speeds; at lower speeds the back pressure in auxiliary exhaust might be lowered, thus somewhat reducing the steam used by the auxiliaries.

Use of Auxiliary Exhaust in Port.—It is probable that the auxiliary exhaust might be used as primary steam for operating evaporators in port. Under these conditions evaporators would be operated in single effect with vapor led to distillers. Should the auxiliary exhaust be insufficient to get the required capacity, one evaporator might be operated at H.P., the vapor being led to coils of the others. The use of auxiliary exhaust for evaporators is employed in the British Navy, and it is reported that 1 pound of exhaust steam will evaporate .8 pound of water, being about the same result that is obtained with single-effect evaporation, using boiler steam reduced to 40 to 60 pounds working pressure.

One objection to the use of auxiliary exhaust for evaporating is that the coils are likely to deposit oil contained in the exhaust steam, and this will get into vapor in case evaporators are afterwards used in double effect. This may, however, be avoided by running drain of L.P. coil to feed tank till oil is cleared out.

CONDENSING APPARATUS.

The advent of the turbine has brought forth the need for higher vacuum, and this has stimulated inquiry into the question of air pumps and condensing apparatus. The ordinary condenser with a single-acting bucket air pump, when well constructed, is capable of giving a vacuum of 27 to 28 inches of mercury; now, when more than this is demanded, special devices are resorted to. The search after a higher vacuum has also brought to the attention of engineers the bad effect of air in condensers, and has developed special designs of condensers in which the exhaust steam and the water are made to pass in reference to each other so that the best

condensing effect is obtained from a given surface, and the condenser is so shaped and the exhaust so conducted that all the surface will become effective, and the deleterious effect of air avoided.

The Alberger dry-vacuum system is one of the results of the investigations. Weir's Uniflex condenser and the Contraflo and counter-current condensers are others. Various other arrangements of baffles and shapes of condensers and disposition of tubes have also been made, with a view of improvement.

The Contraflo condenser, which appears to be quite advisable in principle, however, necessitates a rather elaborate baffling, which is objectionable. The Uniflex condenser embodies the principle of giving the vapor an unrestricted access to the cooling tubes, presenting a relatively large area of cooling surface to the exhaust in its entrance, a gradual decrease in volume of space as the vapor is condensed, by shape of shell and by decreasing pitch of tubes, and causing the exhaust to be distributed over the whole length.

In the Alberger dry-vacuum system, as proposed for naval work, the exhaust vapor is led in at the bottom or side of the condenser and the dry-vacuum pump suction at the top, or side opposite to the exhaust inlet. This should have the tendency to displace the layer of air tending to form at the bottom of the condenser, and will cause the air to be kept in motion and drawn off by the dry-air pump. By this arrangement the condensed water is also kept as warm as possible and off the tubes, since most of the vapor passes only a short way among the tubes, whereas, with the exhaust opening at the top of the condenser, the condensed vapor passes through the entire tube nest. This prevents any considerable portion of the condensing surface from becoming surrounded by air and thus rendered ineffective. The tubes near exhaust opening have a wider space, to allow for vapor volume.

The rotary air pump, of which the Leblanc (described in the November, 1909, number of the JOURNAL), being built by the Westinghouse Company, is a prominent type, and is

urged as having special advantage in lightness, and, as no pistons or pump valves are used, it is thought that it will require very little attention and overhaul. A difficulty is met in this system when it is desired to have the overboard discharge from pump below the water line, since there would be danger of the salt water from water seal backing into the hotwell when the pump stops. It has been proposed to fit a special check valve to prevent the backing up of water left in pipe, and to lead the discharge from the pump above the water line. It is thought that by the successful application of the Leblanc pump considerable space and weight may be saved, as well as there being less need for constant overhaul. This system for naval purposes is, however, yet in an experimental stage, though an installation of this character has been made on a French destroyer, on which satisfactory results have been reported. The Rotrex pump (C. H. Wheeler & Co., see November, 1909, JOURNAL, page 1385) is another type of rotary air pump.

The relative advantages of these different systems, it would appear, should be brought out on the questions (1) of efficiency (*i. e.*, securing of the highest vacuum); (2) the weight of the installation; (3) the facility for installation and space required; (4) the facility for operation and repairs and overhaul required; (5) the steam consumption of the apparatus.

It appears that any of the various types of apparatus may be made efficient and a vacuum of about one inch of the barometer can be secured.

As to weight, there is no very great difference, and, as many of the systems proposed are still in a state of development, exact comparisons of weight are not yet at hand. The question of weight will, however, be a large deciding factor, and if any of the competing systems can demonstrate any considerable saving in weight advantage is likely to lay with them.

As to item (3), the augmentor condenser appears to give very good facility for installation, since room somewhere under the condenser can be found for the augmentor. The augmentor also presents the greatest facility for operation,

since it requires no attendance. In the case of the dry-vacuum system there is both the dry-vacuum and the wet-vacuum pump requiring attention. The augmentor may be applied to a condenser, utilizing the Contraflo, counter-current, Uniflex or other special points of advantage.

As to steam consumption, it is probable that the augmentor system will be more expensive in steam, but the amount of steam used to operate the air-pump system is extremely small, in any of the cases, so that this slight increase in steam consumption for this service is really not very important.

The point about the augmentor is that it does the work, requires practically no attention or overhaul, is reliable, and its cost of installation is not great. The dry-vacuum apparatus introduces some additional auxiliaries which require care and attention. The dry-vacuum installation on the Scout Cruiser *Salem* (November, 1908, JOURNAL, page 979) has been quite satisfactory, though rather heavy.

The dual air pump (Weir), which also employs the dry-vacuum idea in connection with the Uniflex condenser, has as yet not been thoroughly tried out in extended service, though very largely applied in the British service. If it gives the same vacuum that the augmentor will furnish, and the pump is in all respects reliable, it will probably have the advantage of lightness and economy in steam consumption, and thus may prove to be superior to the augmentor.

In view of present information, however, it would appear that condensers built to utilize Contraflo and Uniflex or counter-current principles, combined with an augmentor condenser and single-acting wet-air pump, is the installation likely to give the best satisfaction for naval use at the present time. For highly-powered large vessels, where space and weight are adequate, some of the dry-vacuum systems may work out to advantage, and give better economy. The question of providing adequate volume for exhaust entrance and proper passage for the vapor among the tubes near its entrance is most important. Overbaffling of condensers is in general to be avoided.

It would appear that considerable saving in the size and weight of condensers could be made by employing corrugated instead of plain tubes. The corrugated tubes would be about one-third more efficient, give greater surface in the same space, and the corrugated type can be expanded in the tube sheet, thus doing away with the necessity for packing and glands. The increased cost of corrugated tubes over plain tubes would have to be taken under consideration. The use of glands can also be avoided by having the tubes bent to the segment of a circle—a Normand practice used on Destroyers *Flusser* and *Reid.*

PUMPS.

The principal new development in this field is the introduction of turbine or motor-driven centrifugal pumps for boiler feed and general pumping service.

It is not unlikely that for large power the turbine-driven centrifugal feed pump will offer a very considerable improvement over the reciprocating feed pump at present used, where valve and packing troubles are ever present, and where the steam consumption per power delivered is very great. Tests have been made with some of the best types of reciprocating feed pumps, and the water per horsepower per hour is something like 60 to 150 pounds.

The centrifugal pump has no valves, steam-valve gear or piston packing, and nothing to overhaul except the bearings. The steam consumption is much less, the pump gives a steady pressure, not the intermittent shock. Hence the wear and tear on the piping and the actual resistance in the piping is much less. The pump is fitted with regulating devices by which any pressure or any desired per cent. of excess feed pressure over steam pressure may be obtained.

The principal objection to the centrifugal pump appears to be the extra floor space that is required. It is desirable to have the shaft horizontal, and, thus arranged, the floor space is considerably more than is required for the vertical-piston pump.

A type of feed pump of this character is illustrated in Figs. 6 and 7. This pump can be arranged with three or more stages. It operates at about 2,500 revolutions. The steam consumption is 35 pounds per H.P. and 60 to 70 pounds per water horsepower, exhausting at atmosphere pressure.

Our present piston-feed pumps may be greatly improved by fitting them with automatic regulating valves, by means of which a certain excess feed pressure above steam pressure is maintained. One type of this apparatus is the Foster regulator, described on page 1058 of the August, 1909, issue of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS. Another type of this apparatus is shown in accompanying Fig. 8.

This regulator keeps the pump going at all times only as fast as may be necessary to maintain the required feed pressure, slowing it down or speeding up, as necessary. It thus effectively prevents excessive discharge pressures and thus greatly reduces the strains on pump, piping, feed heaters, etc. As ordinarily used, the feed pumps at full power have their relief valves lifting for a larger part of the time, and this, of course, means a large waste of power. The use of such a pump regulator also makes the attendance on the pump and tending water much easier.

Fire pumps should also be fitted with regulating throttles, which would prevent more than a certain pressure being brought on the discharge. Now dependence is placed on relief valves and hand attendance. The regulator would give the desired pressure in the fire main, no matter whether one or several cocks were open; and if all should be shut off for any reason, it would stop the pump and thus prevent any accident caused by the pressure being increased on pump and piping.

Circulating Pumps.—Compound piston engines for circulating-pump drive have proven very satisfactory, and it is hardly necessary to look for any material change. The use of turbine or motor-driven pumps has been proposed for this service, but it does not appear to offer any particular advan-

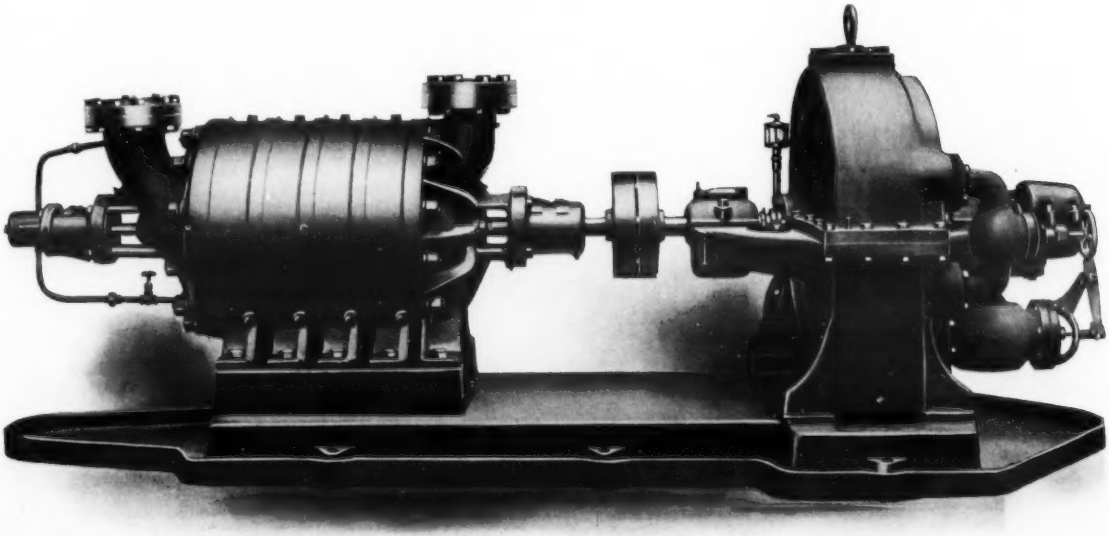


Fig. 6.—Five Stage Turbine Boiler-Feed Pump with Steam Turbine.

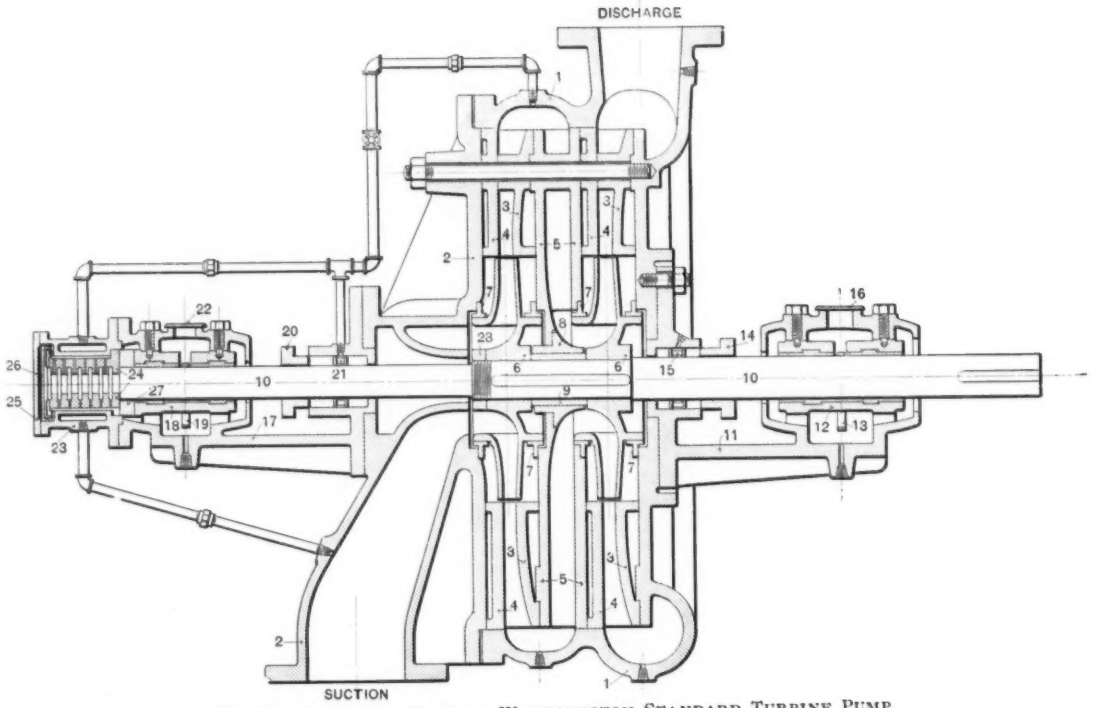


Fig. 7.—Sectional View of Worthington Standard Turbine Pump.

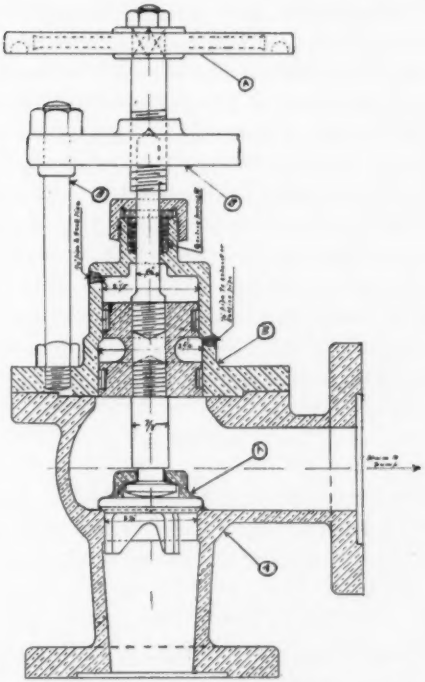


Fig. 8.—Automatic Regulating Throttle Valve for Feed Pump.

tage. If a vessel were operated at a certain constant power, where the speed for the pump was the same all the time, a turbine or motor-driven pump would offer some advantages, but, as it is, the circulating pump is operated but seldom at its designed power. Hence, the piston pump is more economical, since the steam consumption for the turbine is very high when operated at reduced speed.

FORCED-DRAFT BLOWERS.

The use of electric motors for forced-draft blowers appears to give the most reliable installation and one very readily controlled. The principal objections to the electric drive are

that it requires somewhat more weight (considering the additional generators that are required) and it is also rather uneconomical when the blowers are operated at reduced speeds. With the improvement now being made in small steam turbines it is likely that a thoroughly satisfactory and reliable blower drive can be secured by turbines on less weight and with better steam economy than with the electric drive. The electric drive is more easily installed, requires no piping, needs less attention, is more easily operated, and distant speed control from fireroom can readily be accomplished. Such speed control from fireroom can also be secured with the turbine drive. For destroyers and torpedo boats steam turbines would be employed, owing to great need for saving weight.

DYNAMO PLANT.

Turbo-generators are being used exclusively on all new naval vessels, it being usual to supply two to eight units of the same size. The ordinary lighting and power service on a battleship will require 75 to 100 kw., and, owing to this fact, it would appear more advisable to install two smaller units of about 100 kw. each with the others instead of making the installation exclusively of machines of 250 or 300 kw. These large machines, for 80 or 90 per cent. of the time, will operate at below half load, which is at an uneconomical condition. With the smaller units these could be operated at nearly full load with better economy. It would also seem to be very desirable to operate some of the generators by means of oil engines (Diesel). These oil engines are very economical, and it would enable light and power to be on the vessel with all boilers dead. For purposes of repair and overhaul it is often desirable to have all steam off the piping, and with oil-engine dynamo this can be done while still having light, ventilation and power. It might also serve to save coal for a vessel detained at some port or a rendezvous for a considerable time, cases which might happen in war. The installation of several electrically-driven centrifugal fire pumps would enable the

vessel to have practically all her necessary auxiliary services without steam on the ship.

In order to save on watch-standing personnel, and thus have more men available for repair and overhaul, it would be a great advantage to have the smaller dynamo sets, suggested above, placed in the engine room, where the engine-room watch could look out for their operation. The main generators could be shut down and run only when required, and thus nearly all of the present dynamo-room watch-standing personnel would be available for repair and overhaul. The use of oil engines for dynamo drive appears to be specially desirable on torpedo boats where the size of the generator is so small that the turbine gives but poor economy, and requires one of the large main boilers and considerable auxiliary machinery to be in operation.

Another move toward economy that might be used to advantage is to build the turbines divided into a H.P. and L.P. casing; then in port the L.P. turbine could be operated with the auxiliary exhaust steam.

OPERATIONS.

Working economy depends largely on means, methods and conditions of operation, and though the machinery be well designed and well constructed, it may fail in securing good results by being unintelligently and carelessly handled.

The points having a very material bearing on securing economy are in many cases not well understood, or carelessly regarded. Among these the following may be enumerated:

Adjustment of Links.—It may seem strange, but this matter often is entirely misunderstood, and there are cases where engines are operating at one-fifth to one-eighth power with the cut-offs on the H.P. cylinder at practically full gear. This has in some instances been done on trials where there was a penalty for coal consumption.

An increase in the number of expansions increases economy; therefore, at the lower powers, the H.P. cut-off should invariably be run in to cut off earlier, and the full working

pressure kept in the H.P. chest. When the limit of adjustment on the link block is reached still greater linking in can be effected by not moving the reversing shaft all the way over. In fact, some of the reversing gears have a notched quadrant (like those on a locomotive engine) for this very purpose. Also, aside from extra losses in radiation and leaks, the best economy is secured by using steam of the highest temperature.

Feed Heater.—Often the feed temperature is allowed to go away below the 200 or 220 degrees that should be obtained. It usually only requires a little attention to the feed heater to keep this temperature up. The principal causes for improper operation of feed heater are: heater not being properly drained, heater becoming air bound, collection of oil on surface of tubes, heater not fitted with efficient scattering plate so that the feed water passes only through a portion of the tubes, the water in the other tubes lying idle.

Vacuum.—Good vacuum requires the air pump to be in good condition and an absence of air leaks. The principal air leaks are likely to be the piston rods and valve stems of the L.P. cylinders of both main and auxiliary engines, whose exhaust is below atmosphere. Numerous air leaks may also be found in exhaust connections and condensers, as well as air-pump suction piping. Another cause for bad vacuum is that foot valves of air pumps are fitted with seating springs much heavier than necessary; reducing the tension of these springs will, in many cases, raise the vacuum a half inch or more. With turbines vacuum may be materially lowered, due to air leaks through drains and bad air leaks through the L.P. glands.

Use of Too Many Pumps and Auxiliaries.—In many cases too many pumps are kept in operation for best economy. This may apply to feed pumps, where often several pumps are kept going when one pump can easily pump all the feed needed. Bilge and fire pumps are often operated while not being especially needed, or pumps and auxiliaries are operating a greater power than the immediate needs call for.

Steam Leaks.—Loss due to partially closed or leaky valves in drain lines may be very considerable. This loss is shown by excessive temperature in feed tank, when drains discharge to tank. When the drains discharge to condensers there is no way of telling what this loss is. Careful examination and overhaul of drain valves, by-pass valves on traps, etc., appears to be the cure for this evil. Inefficient steam traps or traps not in proper working condition are responsible for a great deal of heat losses of this kind.

Tightness of Pistons and Valves.—This is, perhaps, the most important factor influencing economy of the engines themselves, and requires constant examination and adjustment of these parts. Both valve and piston rings must be kept carefully adjusted so that proper degree of tightness is maintained. The fact that the engine may run smoothly when these various leaks are present, and that many of them can only be detected by considerable investigation and examination, creates a tendency to allow these leaks to go on without remedy till they are of considerable magnitude. One of the greatest helps in keeping these parts in proper condition is the use of graphite for lubricating the sliding surfaces. The tightness of pistons and valves of the auxiliaries must be looked after, as well as those of the main engines, and no steam cylinder should be considered too small and unimportant to go to the trouble of keeping it tight. All piston valves of any appreciable size should be fitted with snap rings, as leaking plug valves are a source of great loss.

Firing.—A good boiler with good firing will give an efficiency of about 80 per cent.; with poor firing the efficiency may fall as low as 50 per cent.; so it is plain to see that firing is a main point in the battle for economy. Proper firing consists of more than throwing coal into the furnace, using the slicing bar and cleaning fires. These are but the rough outline. The main thing, however, is to thoroughly burn every bit of coal put into the furnace with the least excess of air. This is to be secured by uniform and systematic and intelligent firing.

One of the principal aids in securing systematic firing is a reliable firing device; but if the firemen are not efficient or intelligent enough to understand what the object of the device is, it alone cannot bring results. For instance, a fireman might continue to place a certain number of shovels of coal on each time, whether the fire was burning this amount or not, or he might not put on enough. He might not spread it properly, or he might allow clinkers to form. The timing device will serve to assist the firemen in securing uniformity in his operations and help to secure the condition whereby each furnace and each boiler will do approximately the same work, which is the condition that will afford the highest efficiency. Without uniformity in the output of different boilers the greatest efficiency cannot be reached.

Regulating the Draft.—Forcing large amounts of excess air into the furnace leads to waste, since all the excess air has to be heated; hence it should be seen to that a higher air pressure than is neccessary to properly burn the coal used, is not carried. Knowing the capacity of the blowers at different revolutions, it can be determined how many revolutions are necessary to furnish the air needed for the proper combustion at the rate of coal consumption set. Then, if there is difficulty in burning this amount of coal, either the boilers are dirty or defective, or the firemen at fault in some way.

It is very likely that a great amount of excess air is supplied most of the time. Careful attention to this matter can undoubtedly result in a very material saving.

The CO_2 recording apparatus serves to indicate incomplete combustion, and will measure the efficiency of the firemen, to a great extent. It will also indicate the excess of air. It will show when good or bad results are present, and will tell the supervising force when to look for trouble or defective conditions. One of the principal points it will indicate is the amount of excess air. The excess air may come in through the ash pit or it may leak in through the boiler casing above the grates. In either case there is a heat loss, since the extra air is heated up to the temperature of the other gases. The

air leaking in through the casing also has a very deleterious effect in reducing the draft.

It is not unlikely that in many cases but little attention is paid to the tightness of the boiler casing, and here is a point where the gas analysis comes in handy. Attention to leaky casing and uptake doors, tightness of dampers for boilers not in use and leaks in the casing joints will account for many lost heat units which it is possible to save. The efficiency of the boiler may frequently be increased 5 to 10 per cent. by the reduction of excess air and the stopping of air leaks in the casing.

Cleaning Fires.—Considerable coal is lost in careless cleaning for fires. Often large quantities of good fuel are hauled out and sent up with the ashes. This may be avoided to a great extent by seeing that the fire to be cleaned is properly burnt down, that good fuel is winged to one side before the clinker is hauled, by keeping fires well broken up so that the need for frequent cleaning is reduced, and limiting cleaning of fires to the real need for this. It is frequently the case that fires are cleaned once in so often without regard to the rate of combustion or real necessity for cleaning.

The work for improvement and economy lies in all fields. All the personnel, the designer, the builder and the operator can each influence the result to a very great extent, and hence it may be seen that each and every one connected with any of the engineering operations may be a very powerful factor in securing improvement. Each one in his particular sphere must use his means and knowledge to the best advantage.

Poor design or workmanship may be improved by good means and methods of operation, as well as will operative results be improved by excellence in design and construction. On the whole, there must be a community of interest, a thorough understanding and coöperation between the designing, the building and the operating forces. Happily we have in our naval service conditions tending to produce these advantages to a better degree than those of other services.

This close association and community of interest and knowledge must be secured and maintained if the best naval engineering results are to be realized. If by any chance this coöperation and mutual understanding is weakened our engineering efficiency and improvement will suffer.

HEAT TRANSMISSION AND TRANSMITTERS.

By Captain A. B. Willits, U. S. N., Member.

In contemplating the various devices for improving the efficiency of "Heaters," in the broad definition of that term, we are impressed with the fact that most of the efforts thus far made have not been directed towards meeting in any positive manner the physical requirements of heating of fluids, which bar "conduction" and involve the seriatum contact of each molecule with the heat-transmitting surfaces.

In the Scotch boiler a forced circulation of the contained water is set up by special feed directing or by injector circulators, etc., to prevent "dead" pockets and insure the passage of all portions of the liquid mass either over the furnaces or between the hot tubes. This aids the natural circulation induced by differences of temperature, but leaves the actual evaporation to gravity displacement of evaporated molecules by the cooler liquid ones, and necessarily much of the water is repeatedly "circulated" before it is vaporized.

In water-tube boilers a higher effect was obtained through securing the direct passage of the entire mass, in small cylindrical volumes, through hot tubes placed either vertically or so inclined as to facilitate the escape of vaporized molecules and reduce the distance of gravity displacement, and while by increasing the number of the tubes and reducing their diameter, efficiency was increased, it was not found possible to so proportion this form, within practical requirements, as to fully accommodate the slow process of natural convection and prevent the passage of a "core" within these liquid cylinders which failed to absorb its quota of heat by actual surface contact.

With an entirely forced direction of flow and through very small tubes, such as were used in the Thornycroft launch

boilers, the entire feed water may, however, be evaporated in a single passage through the system, by reason of extreme length of travel (in comparison with the diameter) of the liquid cylinder. Such an apparatus is obviously not practical as a marine unit except for very small powers, and, for marine work embodied in heaters, distillers, evaporators or coolers, the larger plain-tube construction, in coil or straight nest form, has, until recently, been found to best meet the requirements.

A departure from the common practice, and involving a slight change in principle, was inaugurated by the Niclausse boiler design. Here the water is directed over (within) the plain tube-heating surface in a film form by leading it first to near the closed ends of the heating tubes through inserted "core" tubes and thus enforcing its return through the annular space between the core tubes and heating tubes.

Theoretically this was a step in advance over the common large-tube water-tube boiler, but practically it has not as yet succeeded in arousing any great enthusiasm among marine engineers in this country. Indeed, the device has no particular feature of merit toward producing a forced convection, for, within practical limits, no matter how thin the film nor how small the diameter of the tubes of plain-tube "heaters," just so long as the water travels along a plain heating surface its heating is accomplished mainly through gravity displacement of the hotter by the cooler particles.

With this in view, and in the belief that a vastly greater efficiency could be secured by thoroughly and continuously breaking up the molecular arrangement of the fluid current by numerous deflections, and thus (by producing a forced convection in lieu of gravity displacement) compel the repeated contact of every molecule with the heating surface in a single passage; and also, with the purpose of increasing the heating surface as far as possible, without excessive friction, per axial foot of travel, a well-known American engineer in civil life has invented and put upon the market a device which marks a long stride towards perfecting "Heaters."

This invention has such obvious and remarkable merit as to appeal at once to all who are keenly interested in heat transmission. Its field of application appears to be so wide and its efficiency and economy so superior to the devices in common use as to render its success already assured; while for purposes where (as on shipboard) least size and weight of apparatus for a given output is of importance, its advantages are pronounced.

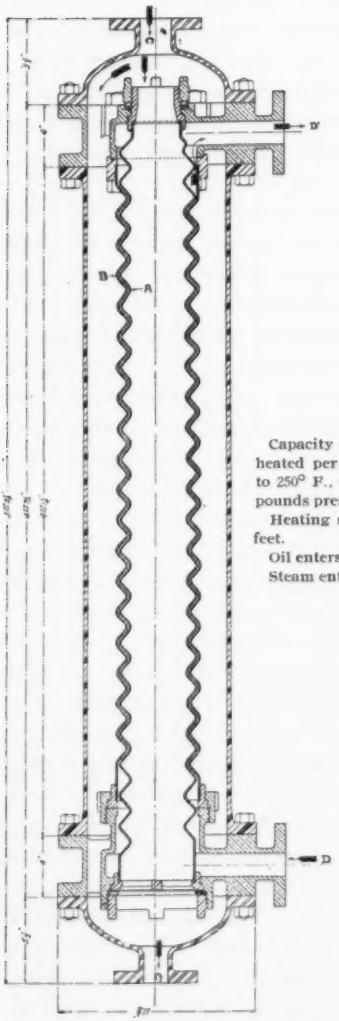
While a number of these devices are authorized for the new ships of our service, and some already installed on destroyers in commission, a brief description of the construction and principle of the apparatus will be interesting to the many who have not yet had an opportunity of examining it.

The figure accompanying these notes shows the heater in its simplest form. Two concentric, spirally-corrugated copper tubes, A and B, form, by the difference in their diameters, a film space between them through which is forced at high velocity the fluid to be heated; while the heating medium embraces both the inner surface of the inner tube and the outer surface of the outer tube, with equal efficiency per square inch of surface. The arrows indicate the direction of flow of both mediums, and but few words are necessary for a complete "reading" of the design.

The prime and essential novelty of this heater lies in the spiral corrugation of the concentric tubes. This construction gives—

First: The desired increased surface over plain tubes per axial foot.

Second: Necessary facility of removal of inner tube for cleaning of film space or replacement of tube with new if change in size of film space is desired, for it is evident that concentric corrugated tubes, if not spiral, could not be used without having the greatest outside diameter of the inner tube slightly less than the smallest inside diameter of the outer tube in order to place one within the other at all, unless the outer tube was made in halves and involved in design and management. This would necessitate far too large a film



Capacity 6,000 pounds of oil heated per hour from 100° F. to 250° F., using steam at 200 pounds pressure.

Heating surface, 10.4 square feet.

Oil enters at D.

Steam enters at C.

"S. AND K." CORRUGATED FILM HEATER.

space as a minimum, as well as interfere with the following and most important feature of economy.

Third: A double effect is provided for breaking up the molecular arrangement of the fluid film and securing the complete forced convection sought for as noted above. This is obvious if one will but follow mentally the film flow. The spiral path deflects constantly that portion of the current which travels along it, and this deflection from a straight line agitates the stream and forces continual change in the position of its particles, which agitation is further enhanced by the conflict of that portion of the fluid which travels axially directly over the corrugations and mixes with the spiral flow. It is difficult to conceive of a simpler or more perfect method for producing a complete and rapid forced contact of all the fluid particles with the heating surfaces without excessive friction, and the results of tests of this apparatus, both as oil heaters and as air coolers, have fully shown that marked efficiency which the student of the design would expect.

From the theory upon which this device is based it is obvious that its advantages are not confined to liquid heaters, but apply to either the heating or cooling of all fluid mediums, gaseous or liquid, wherein the power of "conduction" of heat is so negligible a quantity. The direction of the heat transfer through the transmitting walls of the tubes being immaterial, the use of the apparatus as a cooler is equally convenient.

While the writer has witnessed several interesting tests of this apparatus it is not his purpose to give data at this time, but merely to call attention to the cause of its superior efficiency. Broadly it appears to require but about one-third of the heating surface of a plain-tube type for equal output, and from this some idea of the importance of the invention can be formed.

Of course, like most good things mechanical, it is a patented device and therefore "taboo" to some who fail to realize the important part that the "inventor" plays in engineering progress, and to them an American patent is "double taboo."

But the great majority of engineers feel a keen interest in all inventions of obvious merit, and when these point to meeting urgent needs or to securing marked economies they believe that a prompt investigation and an early adoption of such as "fill the bill" is a sign of good old common sense, which keeps people and nations up to date and puts to the blush that conservatism which prefers the "status quo" to any advance necessitating the toll of a royalty.

THE IMPROVEMENT IN DESIGN OF CONDENSING INSTALLATIONS ON BOARD OF SEA-GOING VESSELS.

By Hans Vogel, Stettin.

Translated by Wm. Wachsman, Associate.

Note.—The following is a translation of an article on the design of marine condensing installations, which appeared in the "Zeitschrift für das Gesamte Turbinwesen," January 30, 1909. Subsequent installments on the same subject appeared in the same paper March 10, 1909, and March 20, 1909. It is proposed to publish extracts from these later papers in the next number of the Journal.

With the rapid increase in speed and engine power of modern ships within the last thirty years the problems presented to the designing engineer in regard to the saving of space and weight became always more difficult. Thus it became necessary to increase the velocity of the water in the pipes; this applied especially to the pipes of the circulating pump. With the introduction of the steam turbine, which demands a very high degree of vacuum and consequently a large amount of circulating water, the problem is of still more importance. Therefore, it appears reasonable to consider the handling of the circulating water somewhat more in detail from the standpoint of the designing engineer.

The handling of this water according to its course is made up of the following four sections:

(*a*) The outboard entrance of the water and the flow through the valves and suction pipes;

(*b*) The passage of the water through the circulating pumps;

(*c*) The flow of the water through the condenser; and

(*d*) The discharge of the hot water overboard.

Before going into details, however, let the more important experiences of the laws of hydromechanics coming into consideration here be briefly summarized.

(1) If water flows through a filled pipe any velocity v in meters per second (without regard to the resistances) corresponds to an expenditure of fall in head, or velocity head,

$$h = \frac{v^2}{2g} \; (g = 9.81 \text{ meters per second per second}).$$

(2) The resistances are independent of the side pressure of the water.

(3) The product of velocity times area must be constant in the entire pipe.

(4) Resistance factors. For straight pipes, according to H. Lang,

$$4\,\xi = \lambda = a + \frac{b}{\sqrt{v \times d}},$$

in which ξ represents the resistance factor, which gives, with respect to a certain area, the existing hydraulic resistance as a part of the velocity head, $\frac{v^2}{2g}$.

This resistance, therefore, is

$$w = \xi \frac{v^2}{2g},$$

d = diameter of pipe, in meters.

Pipes having internal smooth surfaces:

$$a = 0.012, \qquad b = 0.0018;$$

for rough cast-iron and cast-steel pipes,

$$a = 0.02, \qquad b = 0.0018.$$

Example.—If the velocity of the water = 5 meters per second and $d = 0.4$ meter, then in the former case,

$$4\,\xi = \lambda = 0.012 + \frac{0.0018}{\sqrt{5 \times 0.4}} = 0.012 + 0.00128.$$

$\zeta = 0.00332$ for pipes with smooth internal surfaces; in the other case,

$$4\,\zeta = 0.02 + \frac{0.0018}{\sqrt{5 \times 0.4}} = 0.02 + 0.00128.$$

$\zeta = 0.00532$ for unfinished cast pipes.

(5) Gradual central enlargements (See Fig. 1).

$$w = \left(\frac{v_1 - v_2}{2g}\right)^2 \sin\delta = \zeta \frac{v_2^2}{2g};$$

$$\zeta = \left(\frac{F_2}{F_1} - 1\right) \sin\delta.$$

(6) Sections of pipe of circular cross-sectional area for guiding or changing the direction of flow.

Inlet branches according to Figs. 2 to 5.

A close consideration of Figs. 2 and 3 teaches what great resistance a few inexpert edges can call forth in a design. To obtain the simplest castings possible and to minimize the surfaces to be finished, violations are often committed against the most elementary laws.

(7) Pipe bends (according to Fig. 6).

$$\zeta = 0.13 + 0.16 \left(\frac{d}{r}\right)^{3.5}.$$

From this formula the following table is obtained:

$r : d$	2.50	1.67	1.25	1.0	0.835	0.713	0.626	0.556	0.5
ζ	0.14	0.16	0.20	0.3	0.440	0.660	1.100	1.400	2.0

(8) Pipe coils, without change in the radius of curvature, give a similar value for ζ as found for straight piping; for example, heating and cooling coils.

(9) Resistance factors of the valves (without respect to the influence of valve body).

In the following (Table I) let

v = the velocity in meters per second.

ζ = the resistance factor corresponding to v.

The loss in velocity head again is $h = \zeta \frac{v^2}{2g}$.

Table I.—AVERAGE VALUES FOR ξ.

Kind of valve.		ξ
Conical valve with an upper guide and flat lower surface,	Current acting against the lower surface.	1.16
spherical lower surface		0.82
conical-shaped lower surface		0.73
Disk valve, with upper guide		3.00
web guides in seat		5.20
Thin gate valve in a rectangular or circular pipe		0.00
Cock, with rectangular opening through plug		0.00
Flap valve, rectangular or circular, open after turning 90 degrees, with or against the current		0.00

NOTE.—All values apply to valves wide open.

From the foregoing-mentioned formulæ and results of tests the following conclusions can be drawn:

(1) The straightest and smoothest pipe possible should be used.

(2) Drawn and brazed pipes at high water velocities possess only about half the resistance that cast-iron or cast-steel pipes do.

(3) If it should be necessary to use the latter kind of pipe, it is recommended that the same be machined (but only then if the highest efficiency is desired).

(4) It is further advisable to cover the internal surfaces of rough pipe with asphaltum or rubber.

(5) If there should be contractions or enlargements in a line of piping, then the velocity should be gradually decreased or raised by means of conical pieces of as slender a form as possible.

(6) At the inlet places sharp corners are very detrimental (see Figs. 2 and 4). These corners should at least be strongly curved, and, where it is possible, should be made as shown in Fig. 3.

(7) Side outlets (as shown in Fig. 5) always act unfavorably; though if such fittings cannot be avoided, then the angle δ must be kept small.

(8) In pipe bends the radius of curvature should be at least twice the diameter of the pipe, though larger radii of curvature than three times the diameter do not bring any essential gain.

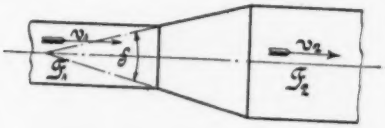


Fig. 1.

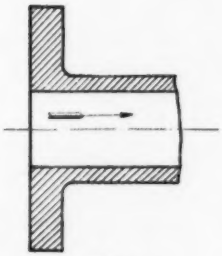


Fig. 2.

Inlet square edge $\xi = 0.5$.
Inlet curved edge $\xi = 0.25$.

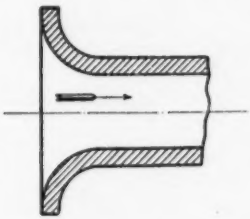


Fig. 3.

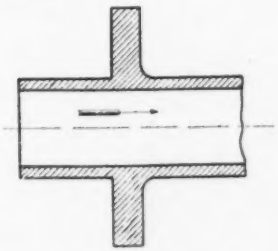


Fig. 4.

Inlet blunt edge $\xi = 0.56$.
Inlet knife edge $\xi = 1.3$.

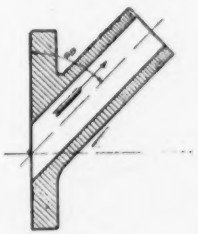


Fig. 5.

$\xi = 0.5 + 0.3 \sin \delta + 0.2 \sin^2 \delta$.

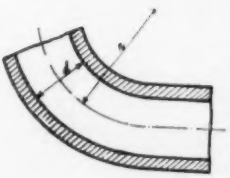


Fig. 6.

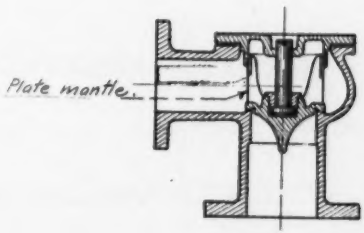


Fig. 7.

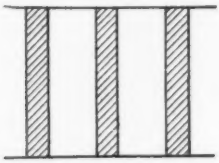


Fig. 8.

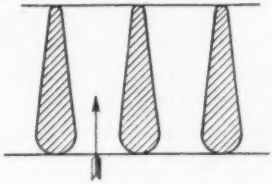


Fig. 9.

Fig. 10.

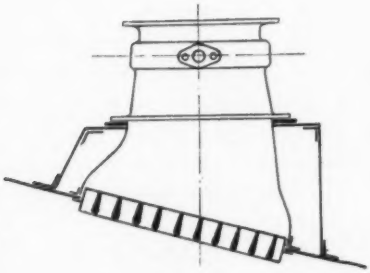


Fig. 11.

Fig. 12.

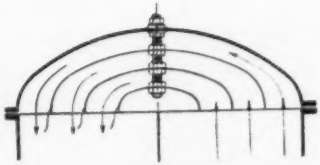


Fig. 13.

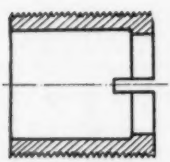


Fig. 14.

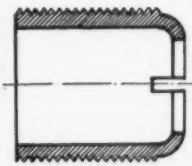


Fig. 15.

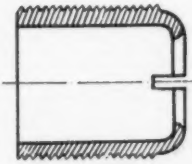


Fig. 16.

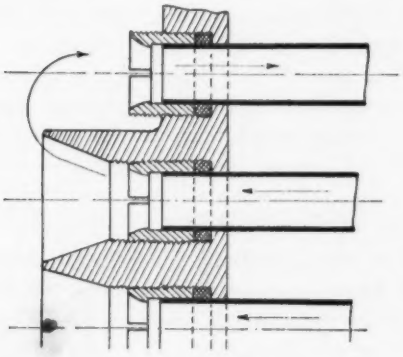


Fig. 17.

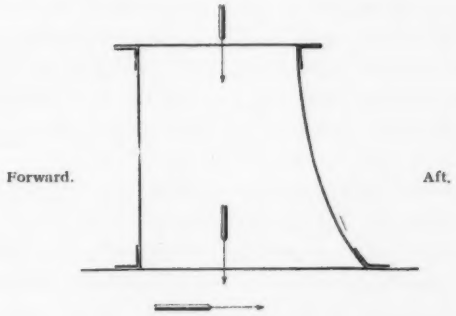


Fig. 18.

(9) Angle and globe valves present a very large resistance, and should be avoided throughout.

The reason for this is that the water in its flow through such a valve must change its velocity as well as its direction at every instant.

If, however, it becomes necessary to use stop valves, then the best efficiency will be obtained from valves having an upper guide and a conical-shaped lower surface (see Fig. 7), and in which the guide ribs are, quite properly, encased by a plate mantle.

(10) Gate valves, cocks and flap valves are the given elements of design for the higher water velocities. The cost of manufacture of this class of valves is, to be sure, somewhat higher than those for angle or globe valves; but in this the gain in work is so considerable that the use of the last-mentioned valves is always to be recommended.

I.—THE CIRCULATING-WATER INLET AND THE SEA CHEST.

The inlet for the circulating water from outboard is generally through a bronze or cast-steel grating. For light vessels some firms make them out of strips of plate. The free area through the grating is made about 1.4 to 2.0 times as large as the area of the suction pipe, the intention being to keep the velocity of flow as low as possible through the grating. The usual cast gratings are very unfavorable for the flow of the water. Mr. Krell (see "Jahrbuch der Schiffbautechnischen Gesellschaft," 1906,) has shown how a grating with a favorable action must be made for air. If such a condition is already laid down for an elastic fluid matter, how much more is a very nearly inelastic fluid entitled to the same.

The shape of the individual grating bars should not be finished as shown in Fig. 8, but as shown in Fig. 9. Gratings according to the latter shape can be cast as well as built up.

In fast vessels the gratings are sometimes permitted to project stepwise outside of the skin. It is believed that by this means the velocity of flow or the pressure through the free surface of the grating is essentially increased. But a large part of the entering water again leaves through the guide pieces curved aft.

The sea chest must be made as small as practicable, so as to reduce the dead weight of the water as much as possible.

It further pays to fit within the watertight sea chest a light framing of plate which will guide the water with the velocity it possesses after passing through the grating gradually over to that in the suction pipe (see Fig. 11). It is generally customary to secure the grating from the outside, and then the guide plate can also be easily fitted in at the same time.

II.—THE LEAD OF THE WATER IN THE PIPING.

As a stopping device against the entering sea water through the sea chest only a sluice or suitable flap valve is to be recommended, for reasons already cited on pages 151 and 152, and in no case should a stop valve be used.

It is best to place the pump as near to the sea chest as possible (a most suitable place is directly over the sluice valve). The suction and discharge pipes should be made as short as practicable and the radii of curvature as large as possible (equal to about twice the diameter of the pipe).

In first-class work the velocity in the piping can amount to 5.5 meters per second.

In consequence of the large variation in area considerable differences in velocity arise during the passage of the circulating water into and out of the condenser. To obtain a favorable action in this case the nozzles on the condenser head should change gradually (see Fig. 12) and the entering and leaving water stream should be in line with the condenser tubes.

Recent practice in general is to use condensers of the counter-current type. In this kind of condenser it would be proper to fit in the rear condenser head two or three zinc strips of the shape shown in Fig. 13, instead of the usual standard zinc plates.

For reasons of strength and economy the endeavor is to make the condenser as long as possible. But this endeavor also seems justified in respect to the flow of the water through the condenser. The condenser ferrules shown in Figs. 14 and 15 produce such a large entering and leaving loss due to contraction that a somewhat higher velocity and a somewhat greater friction of the water does not influence its flow through the condenser tubes as the foregoing mentioned phenomena. To diminish the contraction it is recommended (see Fig. 16) to remove the burr on the condenser ferrules by a reamer or milling cutter (compare also the resistance factors, page 148, and Figs. 2 and 3).

The arrangement with zinc strips, according to Fig. 13, should be given preference to the following one, shown in Fig. 17.

After the circulating water leaves the tubes at the back of the condenser head it must suddenly reduce its velocity considerably. The result is a great loss, due to eddying, which can be essentially reduced by making the back tube sheet as shown in Fig. 17. The increase in weight conditioned by this is inconsiderable, in comparison to the advantages of greater strength, security against the loosening of condenser ferrules and improved lead of the water. The arrangements shown in Figs. 13 and 17 can also be combined.

In most cases the discharge of the circulating water takes place under water. Therefore, the potential energy still contained in the water is lost. If the area at the exit is made at least twice as large as the area of the discharge pipe, and by shaping the discharge pipe as shown in Fig. 18, then a certain proportion of the velocity will be converted into pressure, which will act against the external water pressure. A suction-lift action of the water flowing along the outer skin will likewise occur.

The usual method of securing and packing the tubes by means of screwed ferrules essentially increases the cost of manufacture of condensers. Mr. Praetorius, Naval Constructor, has proposed that the tube be expanded in the tube sheet similar to boiler tubes. The longitudinal expansion of the tubes can be taken care of by bending the tubes a bit in the center and elongating the holes slightly in a horizontal direction in the supporting plates.

Another solution would be to make the tubes straight and to give the condenser shell one or two corrugations similar to a Morison furnace.

The danger of leakage appears to be less in steam-turbine condensers, since here the entrance of the steam is not by shocks, as in condensers for reciprocating engines, but occurs steadily, and therefore the tubes are not so easily given to vibration. If, besides, the inlet for the surplus steam at the exhaust

bend is located as far as possible away from the condenser tubes, leakage is surely less to be feared than, for example, in a Yarrow boiler.

To prevent synchronous vibration of the tubes, due to the vibration of the ship, the distance apart of the supporting plates should be made different in all condensers.

Care must be taken that no air pockets are permitted to be formed in the piping and condenser.

U. S. BATTLESHIP *NORTH DAKOTA.*

DESCRIPTION OF MACHINERY—OFFICIAL TRIALS.

By Chas. B. Edwards,

Superintendent of Machinery, Fore River Shipbuilding Co.

On August 6, 1907, were signed the contracts for two sister ships, the U. S. S. *North Dakota* and *Delaware,* the former to be built by the Fore River Shipbuilding Co., of Quincy, Mass., the latter by the Newport News Shipbuilding and Dry Dock Co., of Newport News, Va.

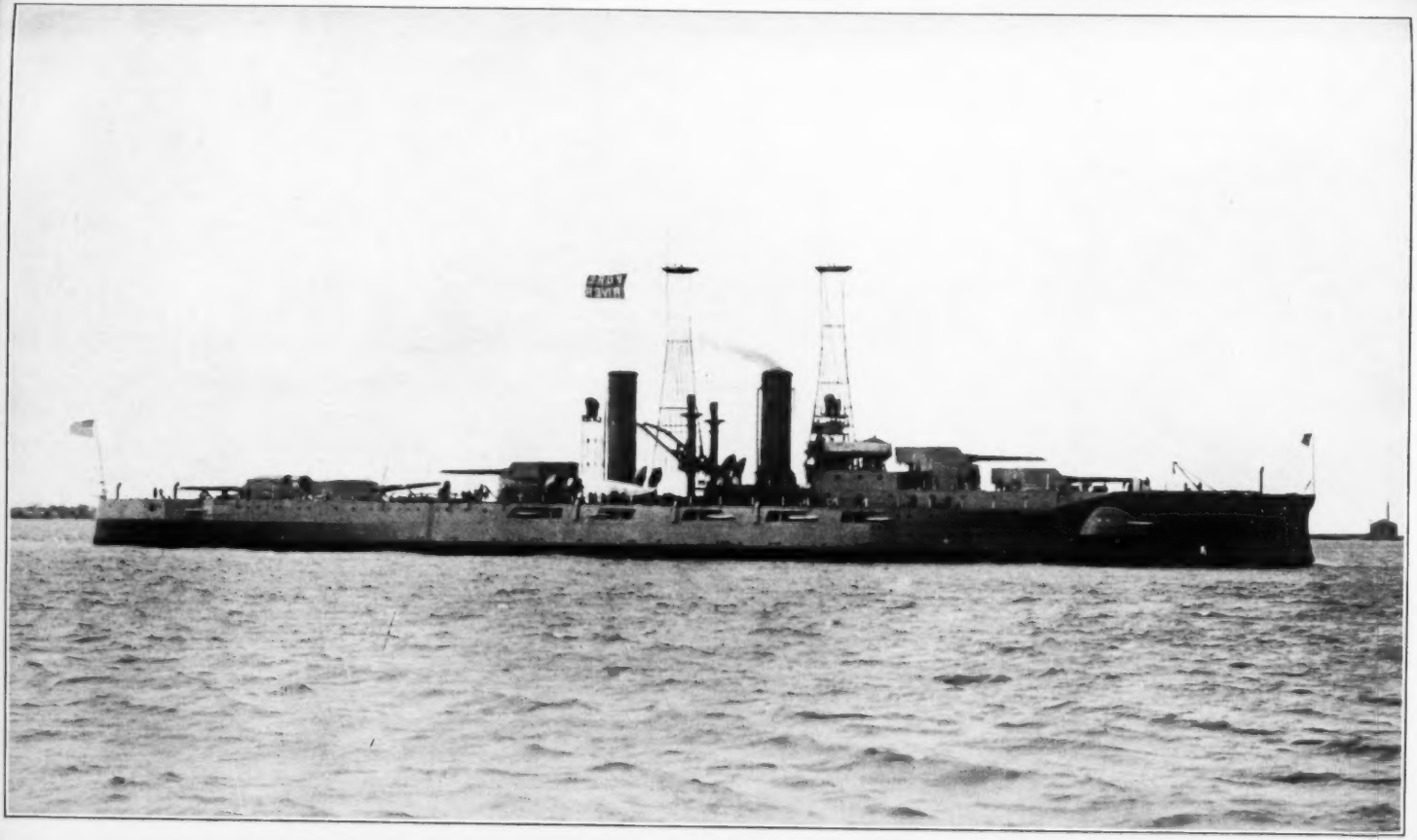
The contract price for the *North Dakota* was $4,377,000, exclusive of armor plates and bolts (except for protective deck), armament and ordnance outfit, and certain articles supplied by the Government. The time of delivery to be on or before the expiration of thirty-four and one-half months from the date of contract.

The following trials are specified in the contract,—

(*a*) A progressive trial over a measured-mile course * * * for standardizing the screws.

(*b*) A full-speed trial of four hours duration * * *. The speed developed by the vessel upon this trial shall not be less than an average of 21 knots an hour * * *, the air pressure shall not exceed an average of 2 inches of water, and the steam pressure in the high-pressure chests shall not exceed 265 pounds above atmosphere.

(*c*) An endurance and coal-consumption trial of twenty-four hours * * *. The speed developed by the vessel upon this trial shall not be less than an average of 19 knots an hour * * *. Careful record shall be made of the coal consumed. All necessary auxiliaries shall be in operation, including all those usually required under service cruising con-



U. S. S. "North Dakota."

ditions, and especially such dynamos as may be necessary for efficiently lighting the vessel and operating at their normal speed the hull-ventilating fans. The evaporating and distilling plant shall be in continuous operation at its normal rated capacity. The pumping and forced-draft air system will also be operated as required.

(*d*) An endurance and coal-consumption trial of twenty-four hours * * * at an average uniform speed of 12 knots * * *. The trial shall be made as nearly as possible under cruising conditions, and the coal consumption for all purposes carefully determined.

The water consumption on trial (*b*) shall not exceed 15.1 pounds; on trial (*c*), 16.1 pounds; and on trial (*d*), 23.2 pounds of water per hour per horsepower, including shaft horsepower of turbines and indicated horsepower of all engineers' auxiliaries in use on the trials, except evaporating and distilling plant.

The following additional items relative to official trials are copied from the precept to the Machinery Trial Board:

"The normal capacity of distilling plant in case of trials (*c*) and (*d*) shall be capacity sufficient to maintain necessary make-up feed and fresh water needed for ship's use during period of trial.

"The water consumption will be determined by means of the water-measuring apparatus installed on board for use during trials. This apparatus is to be operated so that the drain water from feed heaters is to be measured. The discharge from drains, etc., leading to traps can be by-passed. The water from dynamo plant will be by-passed (a prorated portion of this to be charged to forced-draft blowers, to be used in calculating final result as to total water consumption).

"The engineers' auxiliaries, the power of which is to be used for determining water-consumption results, will include the following:

Main circulating pumps.
Main air pumps, wet and dry.
Main feed pumps.

Fire and bilge pumps in continuous use during trial.
Lubricating pumps.
Auxiliary feed pumps in continuous use during trial.
Ice machines.
Power of forced-draft blowers prorated from input during trial."

PRINCIPAL HULL DIMENSIONS.

Length on normal load-water line, feet and inches	510–02 ½
between perpendiculars, feet and inches	510–02½
over all, feet and inches	518–11½
Breadth, molded, feet and inches	84–10½
extreme, to outside of armor, feet and inches	85–02⅝
Trial displacement, tons	20,000
Mean draught to bottom of keel, feet and inches	26–10¾
Coefficient of fineness, block	.60
Displacement per inch immersion at normal draught, tons	71.81
Length of fireroom space, feet and inches	128–00
engine-room space, feet and inches	44–00
Capacity of coal bunkers, tons	2,540.6
reserve feed-water compartments, tons	283.2
oil-storage compartments, tons	381.2

MAIN TURBINES.

The main propelling machinery consists of two 144-inch-diameter Curtis turbines, one on each shaft, each complete in itself, with reversing stages in the after end, and each entirely independent of the other. Their normal capacity is 12,500 shaft horsepower each at 245 revolutions per minute, 265 pounds steam-pressure gauge and 50 degrees superheat at the steam chest, and 27 inches vacuum in the exhaust chamber.

Fig. 1 shows a longitudinal section of one of the turbines, which has nine ahead stages, five on separate rotor wheels and four on a drum, and two reverse stages. Steam is admitted through a cast-steel steam chest to twenty first-stage nozzle ports, eighteen of which are supplied with valves operated by a handling gear conveniently located above the operating platform. In order to reduce the first-stage pressure to such an amount that the casing can be made of suitable weight, the

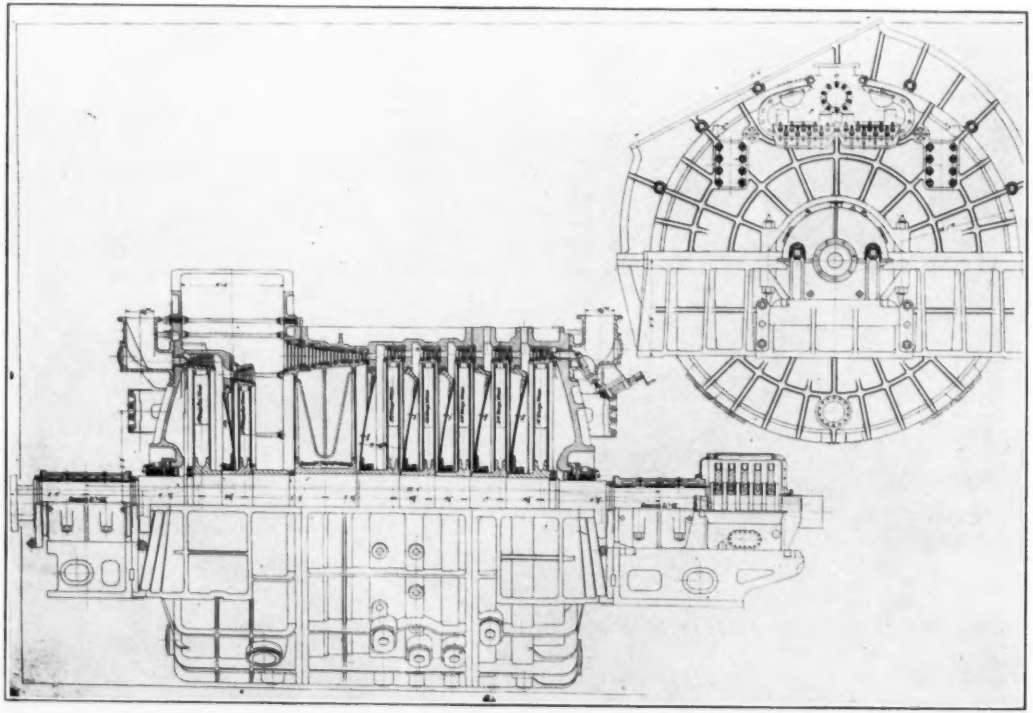


Fig. 1.—SECTION OF MAIN TURBINE (CURTIS), "NORTH DAKOTA."

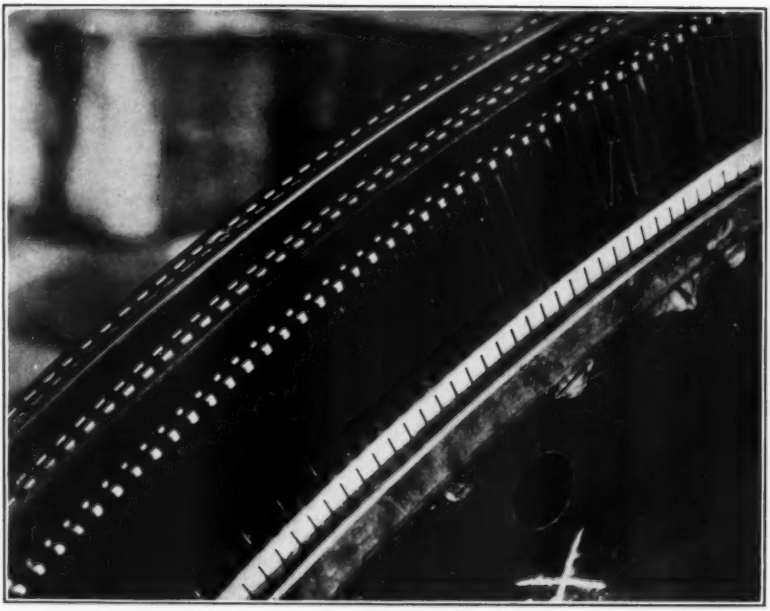


Fig. 2.—BLADING, CURTIS TURBINE.

first stage is designed for a greater pressure and energy drop than the other ahead stages and the nozzles are of the expanding type, the steam passages being machined in a composition casting with a special machine. On leaving the nozzle the steam passes through alternate rows of moving and stationary buckets, which in turn absorb its velocity and reverse its direction, to the receiver in front of the second-stage nozzle ports. The pressure remains the same throughout the stage, the changes in its condition being merely those of velocity and quality. In the remaining ahead stages the nozzles are made with parallel walls by casting nickel-steel plates in cast-iron or composition segments. The arrangement of nozzles and buckets in the drum stages is exactly the same thermodynamically as in the other stages. The part of the circumference through which the nozzles extend gradually increases from a small arc in the first stage to the whole circumference in the sixth and remaining stages. After the steam leaves the last row of buckets in the ninth stage it passes through a rectangular nozzle in the exhaust casing and a built-up steel exhaust pipe to the condenser. For running astern the steam enters the turbine through a similar chest and passes through the two reverse stages into the exhaust chamber, which is common to the ahead and reversing turbine. There are sixteen first-stage reverse nozzles, six of which can be closed. The second-stage reverse nozzles go all the way around the circumference. Both are of the expanding type.

The system of blading (Fig. 2) represents one step in the development of the mechanical features of this type of turbine. The extruded metal buckets are riveted at the proper angle into a channel milled from cold-rolled steel stock. Slots are sawed into the channel, after which is bent around the circumference of the wheel or drum and the legs are caulked into grooves. The intermediates are made in just the same way and are caulked into holders, which in turn are bolted to the inside of the turbine casing.

The rotor wheels and drum are of the built-up type, being made by riveting or bolting steel-plate webs to a cast-steel

spider hub and a cast-steel rim. They are forced on the shaft one after another by hydraulic pressure and keyed.

The diaphragms between the high-pressure stages are made by riveting cast-steel rims and forged-steel hubs to steel plates pressed hot into a spherical shape. A lip is machined on the outside of the rim which fits into a groove on the inner circumference of the turbine shell. The composition bushing inside the hub has machined on its inner surface a V thread, about one-quarter inch deep, plugged at each end. This threaded surface together with the shape of the bushing itself reduces to a minimum not only the leakage of steam between the stages, but also the possibility of any mechanical interference. In the drum stages the inner circumferences of the nozzle segments are similarly machined to perform the same function.

The turbine casing is of cast iron in six parts with one horizontal and two vertical transverse joints. An external drain pipe with a valve leads from a drain pocket in the bottom of each stage to the next lower stage, and the exhaust chamber is drained to the main condenser by means of an ejector. A clearance indicator is fitted to the forward end of the casing to show the relative axial position of moving and stationary parts. The stuffing boxes around the shaft where it passes through the casing are of the carbon type, the leakage of steam or air being prevented by segments of accurately machined carbon rings, which are held against the surfaces of the stuffing box and of the shaft by springs and by the pressure of the steam itself. A drain is provided to a low-pressure stage of the turbine and to the main condenser for steam that leaks past the first rows of carbons, and a steam seal is used to prevent the inward leakage of air when the stuffing box is called upon to maintain a vacuum in the casing.

The main bearings are cylindrical, with composition lower brass and cast-iron pedestal and cap. The thrust block is incorporated with the forward main bearing and is provided with five horseshoes to take whatever unbalanced thrust may result from the action of the propeller and of the steam pressure on

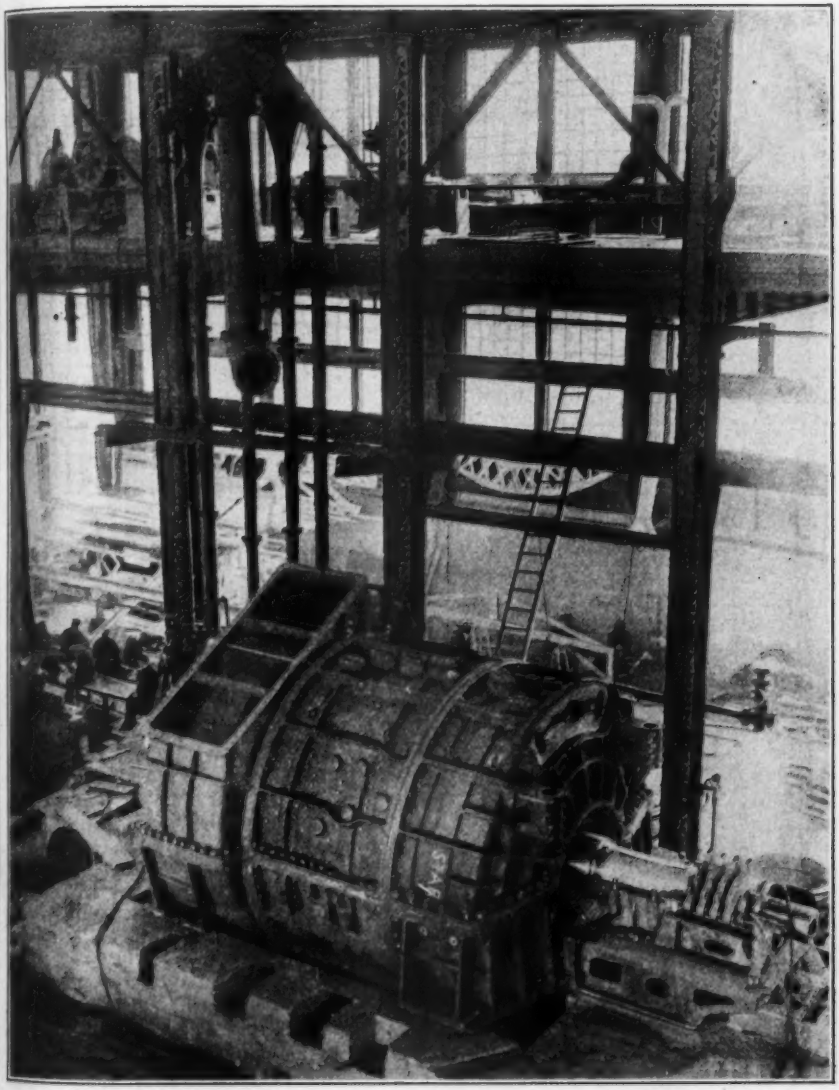


Fig. 3.—Curtis Turbine in Shop.

the forward end of the rotor drum. The main bearing supports are bolted rigidly to the forward and after ends of the turbine shell.

TURBINE NOZZLE AND BUCKET DATA.

.....	Ahead.									Reverse.	
						Drum.					
Stage	1	2	3	4	5	6	7	8	9	1	2
Pitch diameter, inches	144	144	144	144	144	141	141	141	141	135	123
No of nozzle openings	20	56	72	112	240	256	220	220	360	16	70
Radial ht of nozzle mouth, inches	1.4	1.8	1.8	1.8	1.8	1.88	2.90	4.95	7.5	1.4	2.5
No. of rows of moving buckets	4	3	3	3	3	3	3	3	3	5	4
stationary buckets	3	2	2	2	2	2	2	2	2	4	3
Height of buckets, inches—											
1st row, moving	1.68	2.16	2.16	2.16	2 16	2.256	3.48	5 76	8.4	1.7	3.04
2d row, stationary	1.96	2.52	2.52	2 52	2.52	2.632	4.06	6.57	9.3	2.0	3.58
3d row, moving	2.24	2.88	2 88	2.88	2.88	3.008	4.64	7 38	10.2	2 3	4.12
4th row, stationary	2.52	3 24	3.24	3.24	3.24	3.384	5.22	8.19	11.1	2.6	4.66
5th row, moving	2.80	3.60	3.60	3 60	3.60	3.76	5.80	9.0	12.0	2.9	5.20
6th row, stationary	3.08	.....	.....	.....	.....	.....	.....	.....	.....	3.2	5.74
7th row, moving	3.36	.....	.....	.....	.....	.....	.....	.....	.....	3.5	6.28
8th row, stationary	.....	.....	.....	.....	.....	.....	.....	.....	.....	3.8	.....
9th row, moving	.....	.....	.....	.....	.....	.....	.....	.....	.....	4.1	.....
No. of buckets per row—											
1st row, moving	720	720	720	720	720	705	705	705	705	878	800
2d row, stationary	120	160	218	310	600	705	705	705	705	195	800
3d row, moving	720	744	744	744	744	728	728	728	724	878	800
4th row, stationary	120	178	243	346	670	786	786	786	772	195	800
5th row, moving	739	820	820	820	820	802	802	802	802	878	800
6th row, stationary	130	.....	.....	.....	.....	.....	.....	.....	.....	195	800
7th row, moving	820	.....	.....	.....	.....	.....	.....	.....	.....	878	800
8th row, stationary	.....	.....	.....	.....	.....	.....	.....	.....	.....	195	.....
9th row, moving	.....	.....	.....	.....	.....	.....	.....	.....	.....	878	.....

OPERATING GEAR.

Each turbine is operated ahead and reverse by two $13\frac{1}{2}$-inch balanced throttle valves admitting steam from the main steam line to the ahead and reverse steam chests respectively, each entirely independent of the other, and controlled by a hand-wheel from the handling platform. The bodies and valves are cast steel and the seats are monel-metal. A 3-inch by-pass admits steam around the throttle to each chest. Ahead of the two throttle valves in the steam line is an emergency stop valve of the butterfly type operated by hand from the handling platform.

TURNING AND LIFTING GEAR.

To each shaft is fitted a turning gear driven by a 5×5-inch engine. The engine is handled by means of a piston reversing valve operated by a hand lever on top of the valve-chest, and drives through a worm and worm wheel a second worm, which

may be thrown into mesh with a large worm wheel on the after coupling of the main turbine shaft. The ratio of engine turns to shaft turns is about 400 to 1. The shaft may also be turned by means of a hand-ratchet wrench applied to the end of the driving-worm shaft.

A lifting gear is provided for handling the rotor and top half casing during erection or dismantling of the turbine. A 30-horsepower motor with a normal speed of 540 revolutions per minute, located in the engine hatch at the height of the protective deck, acts through a system of worms and worm wheels on a threaded spindle at the forward and after ends of the turbine, which will raise or lower either the top half casing or the rotor, or both, at the rate of about one inch per minute. The lifting yokes are so guided as to prevent damage to the blading either on the rotor or in the shell, and supports are furnished for holding the top half casing in place after it is raised.

SHAFTING.

The single length of line shaft between the main turbine and astern-tube shaft is supported by two spring bearings between which is fitted a Föttinger torsion meter for measuring shaft horsepower. The stern-tube shaft is covered with a watertight composition sleeve throughout the length of the stern tube and at the two stern-tube bearings. These bearings are of composition and are fitted with lignum vitae. The inboard coupling is so made that the shaft can be drawn outboard. The propeller shaft is supported by two strut bearings each fitted with lignum vitae. The shaft has composition sleeves at these bearings.

SHAFT DATA.

Main turbine shaft :	
Distance from center to center of main bearings, ft. and ins.	22–06
Over all length, feet and inches	31–09
Maximum diameter, inches	32
Size of bearings, inches	21×42
Diameter between thrust collars, inches	15
Number of collars	6
Diameter of collars, inches	26

Main turbine shaft:	
Width of collars, inches	2
Space between collars, inches	4
Diameter of bore at collars, inches	7
main bearings, inches	12
between main bearings, inches	19
Line shaft:	
Length, feet and inches	22–08⅞
Outside diameter, inches	14
Inside diameter, inches	8
Size of spring bearings, inches	14½×20
Stern-tube shaft:	
Length, feet and inches	51–03½
Outside diameter, inches	14½
Inside diameter, inches	8
Size of stern-tube bearings, inches	16½×60½
Propeller shaft:	
Length, feet and inches	48–06½
Outside diameter, inches	14½
Inside diameter, inches	6
Size of forward strut bearing, inches	16½×36½
propeller strut bearing, inches	16½×60½

PROPELLERS.

The two propellers are solid castings of monel-metal with three blades turning outward for ahead motion. The driving faces of the blades were machined to a true-screw surface by means of a propeller-planing machine and afterward polished. The backing faces were ground smooth. Each propeller was swung on a mandrel and balanced by taking off excess metal from the backing faces. The hub is taper-bored and keyed to the shaft, and at the forward end is fitted a gland with rubber packing. The nut is securely locked and is covered by a composition cap which is turned and polished outside. The clearance spaces inside are filled with tallow and all recesses on the outside are carefully covered with plates.

PROPELLER DATA.

Number of blades	3
Diameter, feet and inches	13–0
Pitch, feet and inches	10–4
Ratio of pitch to diameter	.795
Developed area, square feet	71.3

Projected area, square feet	64.0
Developed area ÷ disk area	.537
Height of lower tip of blade above keel, feet and inches	1–4⅛
Immersion of upper tip of blade at normal draught, feet and inches	12–6½
Diameter of hub, forward end, inches	28
after end, inches	23

MAIN CONDENSERS.

One cylindrical main condenser is located athwartships across the after end of each engine room. The shell is 8 feet 7½ inches inside diameter, of 7/16-inch steel plate, with two double-butt longitudinal joints and three outside circumferential T-bar stiffeners. The exhaust nozzle is also of steel plate, and is 10 feet by 4 feet 5 inches inside at the vertical flange where it joins the exhaust pipe. This nozzle is strengthened by vertical stiffeners. In the bottom of the shell are two 9½-inch flanged connections for the air-pump suctions.

The water chests are composition ⅝-inch thick. The outboard one is divided by a horizontal partition into a lower injection chamber with a 24-inch-diameter water-inlet nozzle, and an upper discharge chamber with a 24-inch outlet nozzle. Valves in the partition, operated by an outside lever, when open, allow the circulating water to pass overboard direct.

The tube sheets are of rolled composition 1 inch thick. The glands are counterbored at the outer end to prevent the tubes from crawling. Cotton-tape packing is used. Each tube sheet is stiffened by a composition stiffener running across the sheet just below the horizontal diameter, and there are nine longitudinal stays of double extra-strong wrought-iron pipe between the sheets. Screwed into the ends of these stays are stay bolts running through the water chests.

There are five rolled-composition supporting plates ¼-inch thick. In each tube hole is a ferrule ¾-inch long, the inside of which is an easy running fit on the tube, and the ends of which are expanded after they are driven through the plate.

The tubes are of seamless-drawn composition, No. 16 B. W. G., ⅝-inch outside diameter, and 13 feet 2 inches long. They are 1 inch between centers. In that region where the entering

steam strikes the tubes every other row is removed, so that the passage of steam is not hindered. There are 6,904 tubes in each condenser, with a total cooling surface of 14,683 square feet.

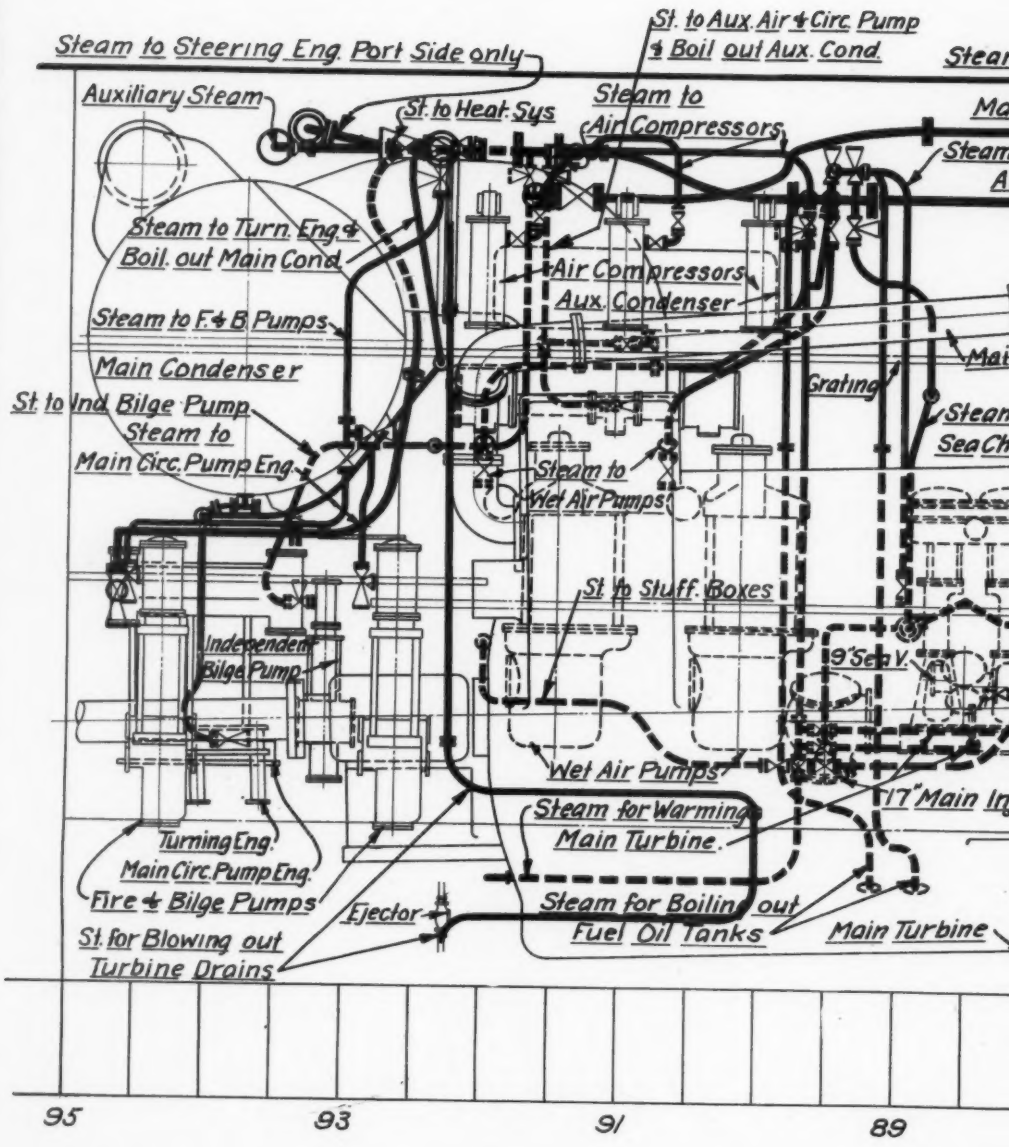
PUMPS.

Main Air Pumps.—In each engine room are two Weir monotype wet-air pumps and one Weir twin rotative dry-air pump. Each wet-air pump has an independent suction from the bottom of the main condenser, and there is a cross connection between the two suction pipes. They discharge to the top of the feed and filter tank. In the wet-air pump suction is an air chamber to which is connected the dry-air pump suction. The dry-air pump discharges to the atmosphere at the main deck.

Main Circulating Pumps.—Water is circulated through each main condenser by a double-inlet centrifugal circulating pump with a capacity of 15,000 gallons of water per minute at revolutions not to exceed 265 per minute. The diameter of the impeller is 4 feet and the shell has 24-inch diameter suction and discharge nozzles. Each pump is driven by an independent compound steam engine, 11 × 22 × 10 inches.

FEED AND FILTER TANK.

In the forward outboard corner of each engine room is a feed and filter tank of ¼-inch steel plate thoroughly galvanized, 10 feet high, 10 feet long and 7 feet 1½ inches wide, containing a filter chamber 30 inches deep in the top and a feed tank proper in the bottom. The filter chamber has a capacity of 938 gallons and the feed tank proper a capacity of 4,360 gallons. In the filter chamber are seven vertical baffle plates between which horizontal perforated plates form three chambers for the Loofa filtering material. The arrangement of baffle plates is such that the filtering material is always submerged and the flow of the water through it is downward. A 7-inch overflow pipe leads from the bottom of the tank up through the top and to the bilge through a funnel, so that any water


Steam to Steering Eng. Port Side only
St. to Aux. Air & Circ. Pump & Boil out Aux. Cond.
Auxiliary Steam
St. to Heat. Sys
Steam to Air Compressors
Steam to Turn. Eng. & Boil. out Main Cond.
Air Compressors Aux. Condenser
Steam to F. & B. Pumps
Main Condenser
St. to Ind. Bilge Pump
Steam to Main Circ. Pump Eng.
Steam to Wet Air Pumps
St. to Stuff. Boxes
Independent Bilge Pump
9" Sea V.
Wet Air Pumps
17" Main In
Steam for Warming Main Turbine
Turning Eng.
Main Circ. Pump Eng.
Fire & Bilge Pumps
Ejector
Steam for Boiling out Fuel Oil Tanks
Main Turbine
St. for Blowing out Turbine Drains
Grating
95
93
91
89

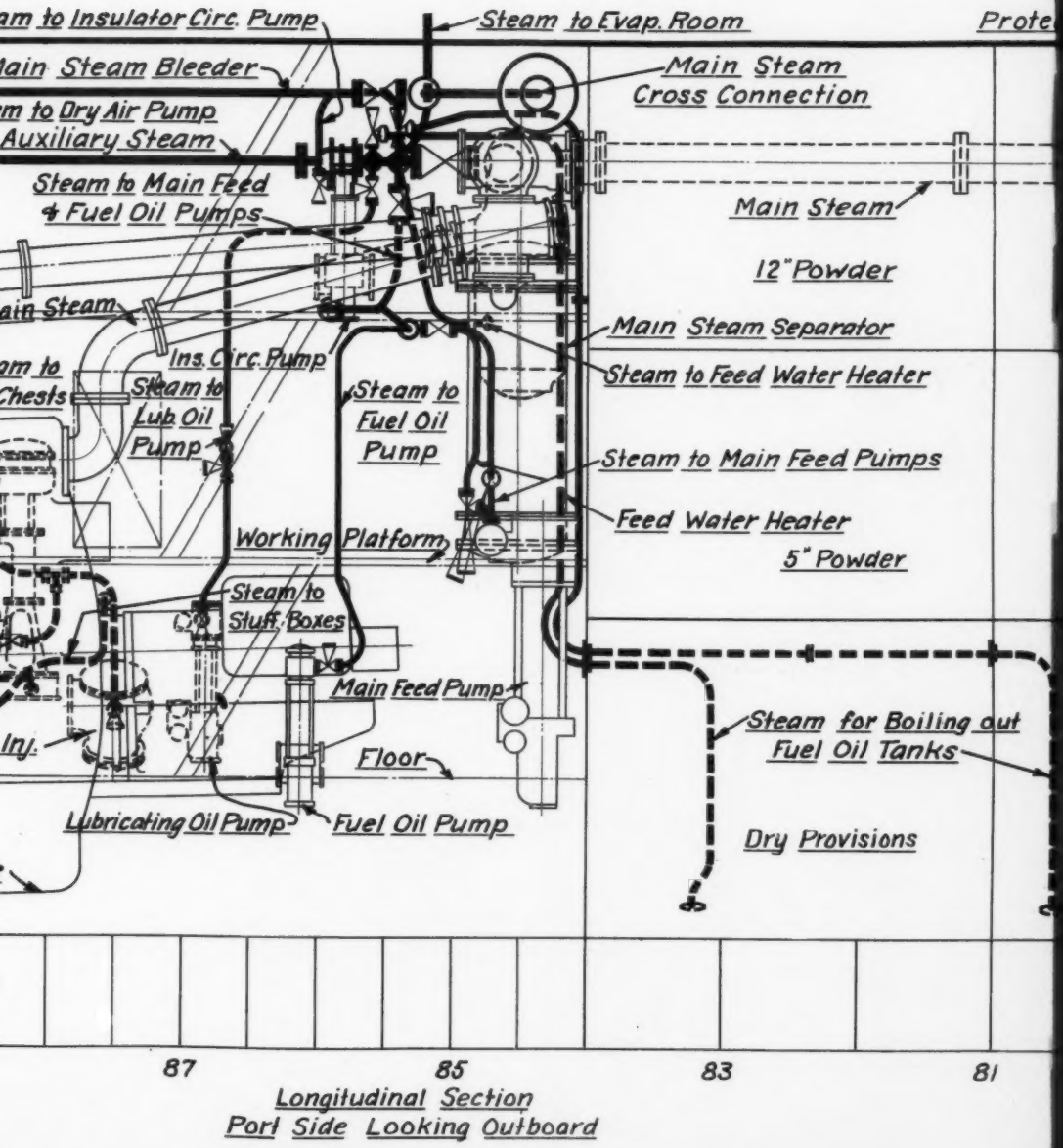
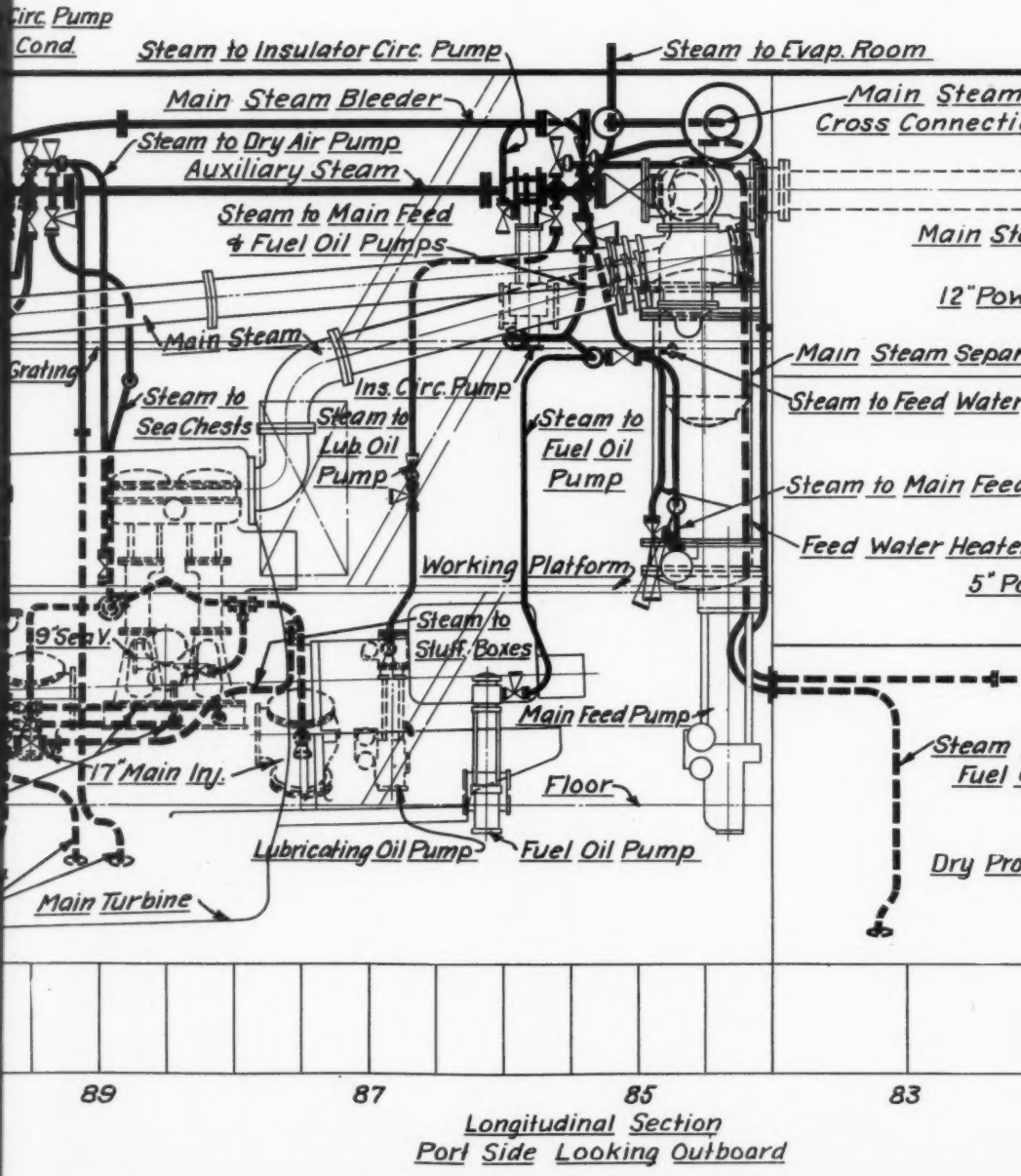
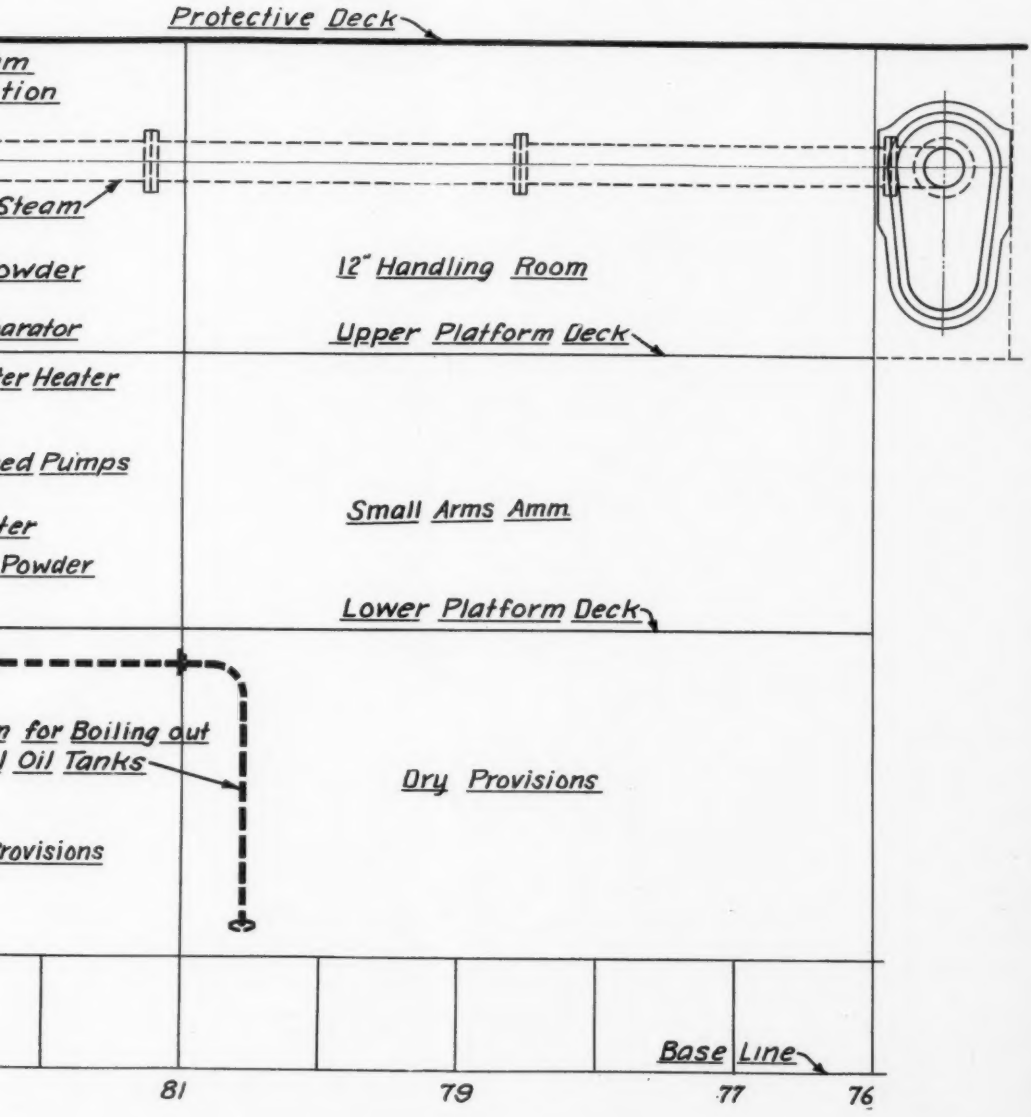


Fig. 4.—Arrangement of Engine Room.



Longitudinal Section
Port Side Looking Outboard

Fig. 4.—Arrangement of Engine Room.


Protective Deck
Steam
owder
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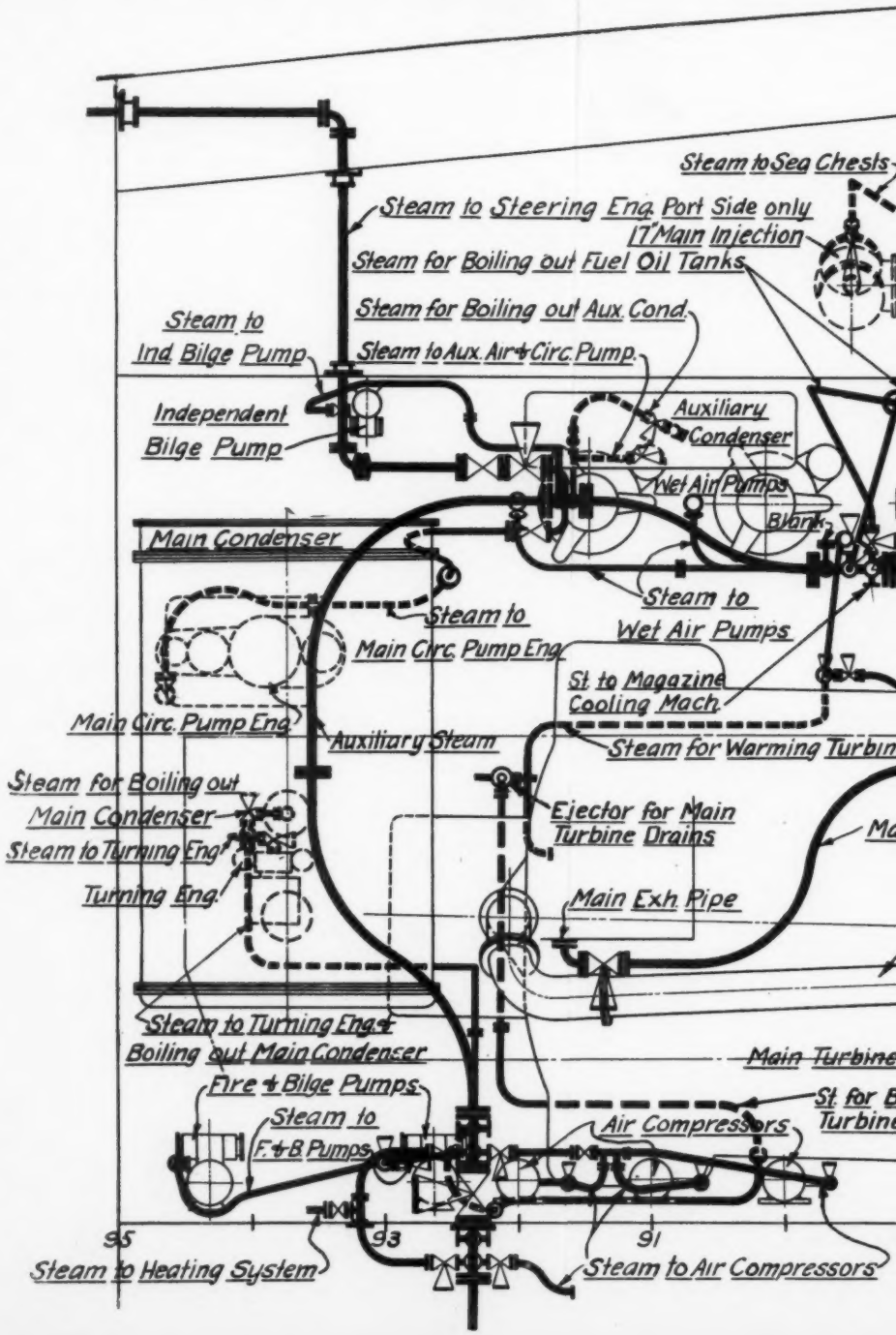
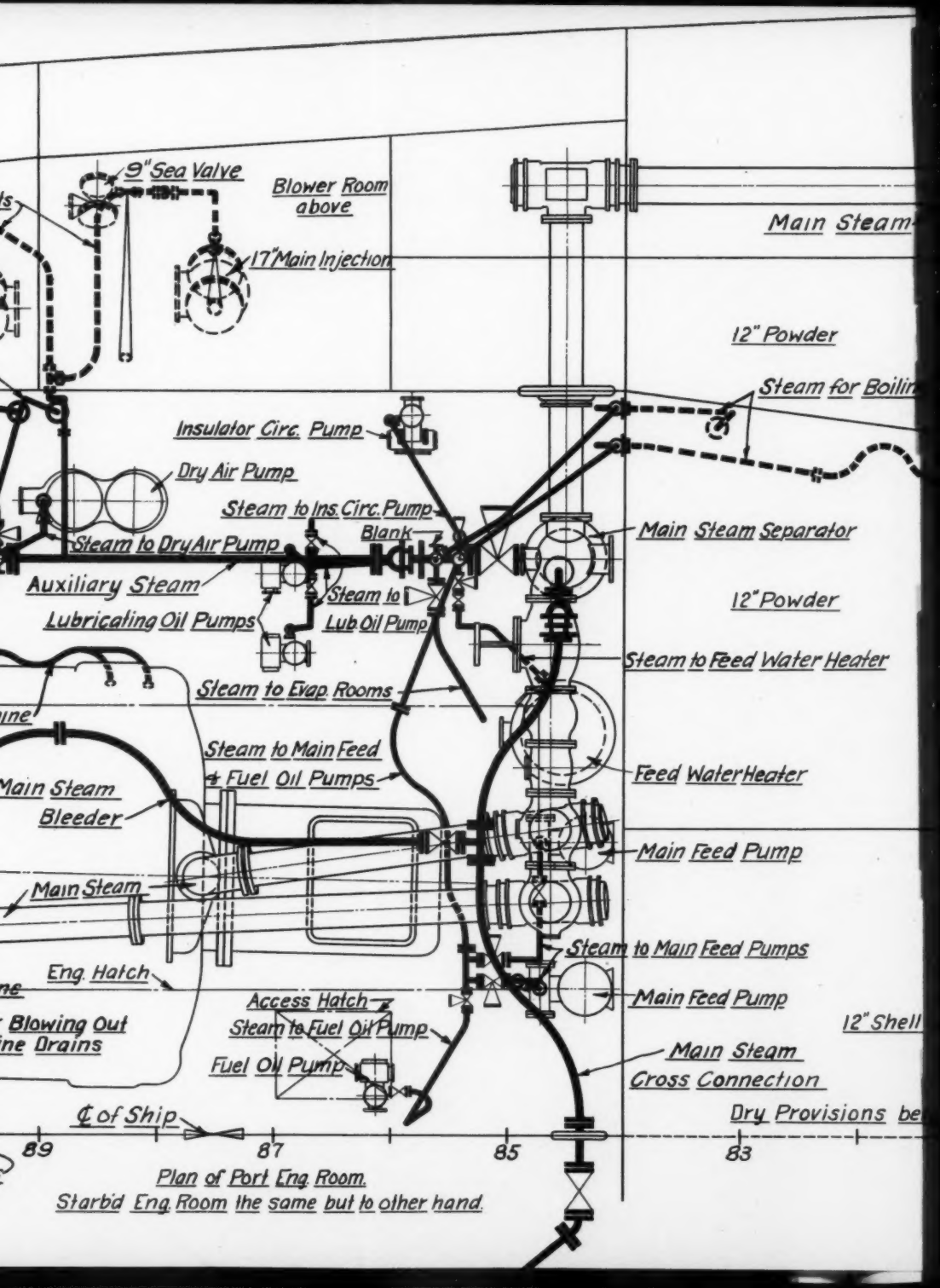
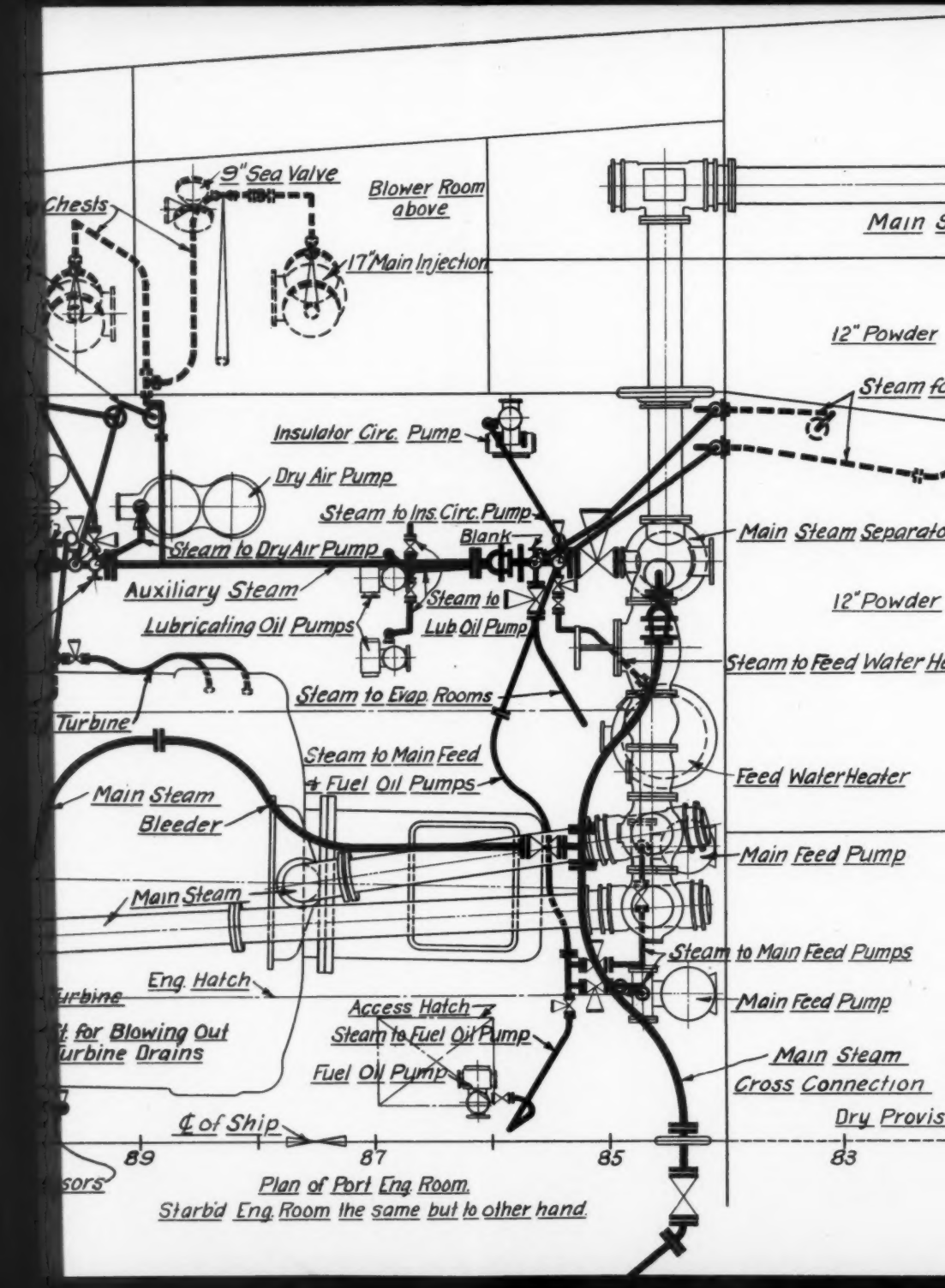
er Heater
ed Pumps
ter
Powder
12" Handling Room
Upper Platform Deck
Small Arms Amm
Lower Platform Deck
n for Boiling out
Oil Tanks
Dry Provisions
rovisions
Base Line
81
79
77
76

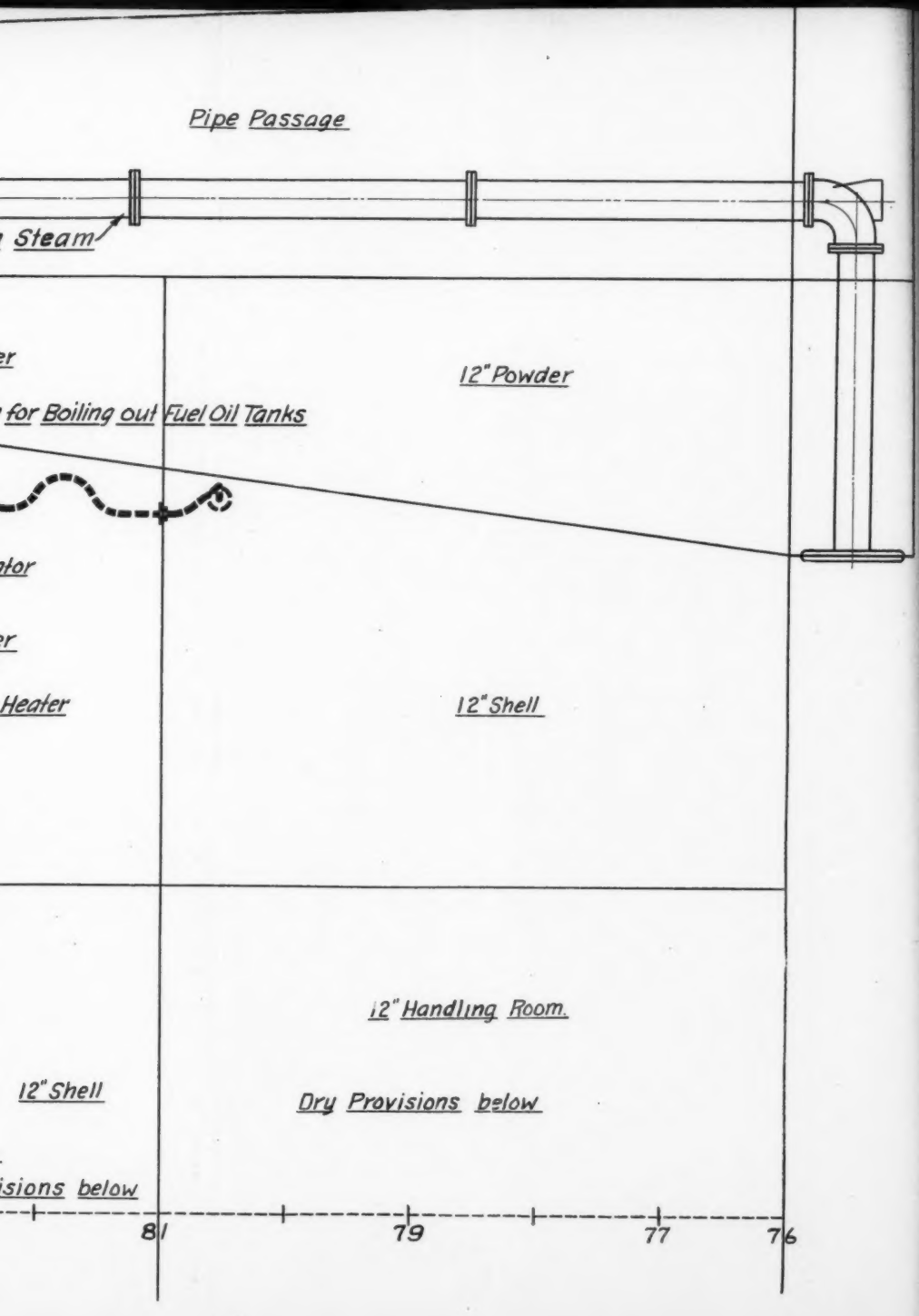

Steam to Sea Chests
Steam to Steering Eng. Port Side only
IT Main Injection
Steam for Boiling out Fuel Oil Tanks
Steam for Boiling out Aux. Cond.
Steam to Ind. Bilge Pump
Steam to Aux. Air & Circ. Pump
Independent Bilge Pump
Auxiliary Condenser
Wet Air Pumps
Bilge
Main Condenser
Steam to Wet Air Pumps
Steam to Main Circ. Pump Eng.
St. to Magazine Cooling Mach.
Main Circ. Pump Eng.
Auxiliary Steam
Steam for Warming Turbin
Steam for Boiling out Main Condenser
Steam to Turning Eng.
Ejector for Main Turbine Drains
Turning Eng.
Main Exh. Pipe
Steam to Turning Eng. & Boiling out Main Condenser
Main Turbine
Fire & Bilge Pumps
St. for Turbine
Steam to F. & B. Pumps
Air Compressors
95
93
91
Steam to Heating System
Steam to Air Compressors



Plan of Port Eng. Room.
Starb'd Eng. Room the same but to other hand.



Plan of Port Eng. Room.
Starb'd Eng. Room the same but to other hand.


Pipe Passage
Steam
12" Powder
for Boiling out Fuel Oil Tanks
Heater
12" Shell
12" Handling Room.
Dry Provisions below
12" Shell
81
79
77
76

passing down it may be seen. On the outside of the tank is fitted a gauge glass and a graduated brass scale for showing the height of the water.

FEED WATER HEATER.

In the feed-pump discharge, near the forward bulkhead in each engine room, is a feed-water heater of the direct-flow type. The shell is of $\frac{3}{8}$-inch steel plate, $33\frac{1}{4}$ inches inside diameter, with a circumferential expansion joint in the middle. The water chests are of cast-steel $1\frac{1}{8}$ inches thick, with 6-inch inlet and discharge nozzles. The tube sheets are of composition, $1\frac{1}{2}$ inches thick; the ends of the tubes are annealed and expanded into them. Each heater has 965 composition tubes, $\frac{5}{8}$ inch outside diameter, No. 16 B.W.G., 15/16-inch pitch, 6 feet 11 inches long, with a total heating surface of 1,050 square feet. Either live or auxiliary-exhaust steam is used for the heating agent. The heater shell drains to the trap-drain main or to the feed and filter tank.

AUXILIARY CONDENSER.

Near the outboard bulkhead of each engine room is an auxiliary condenser supplied with a combined auxiliary air and circulating pump. The condenser has a 7/16-inch steel shell, 27 inches inside diameter. Exhaust steam enters a 7-inch-diameter nozzle through a spring relief valve. The air-pump suction is $5\frac{1}{2}$ inches diameter. The water chests are of composition, 7/16-inch thick, the after one being divided by a horizontal partition into a lower injection chest with a 5-inch-diameter inlet nozzle and an upper discharge chest with a 5-inch-diameter discharge nozzle. The tube sheets are of composition with glands similar to those in the main condenser. Each condenser has 526 composition tubes, $\frac{5}{8}$ inch outside diameter, No. 16 B.W.G., 7 feet $2\frac{1}{8}$ inches long, with a cooling surface of 600 square feet. They are spaced one inch between centers.

FORCED LUBRICATION SYSTEM.

Beside the regular oil system a forced-lubrication system is provided in each engine room, the essential features of which are as follows:

Two piston oil pumps.

One oil-pressure regulator attached to pump steam line.

One Macomb oil strainer in oil-pump discharge.

One 110-gallon gravity oil tank.

One 170-gallon drain and cooling tank.

The pumps take oil from the drain tank and deliver it through the strainers to the gravity tank just under the protective deck, the regulator maintaining a pressure of about 20 pounds in the system. From the gravity tank the oil goes to the main turbine bearings and thrust horseshoes, from which it drains back by gravity to the drain tank.

The cover of the drain tank forms a header from which water-service coils are suspended. These may be easily removed for examination and cleaning. A by-pass around the oil strainer is furnished to allow the strainer basket to be removed and cleaned without stopping the flow of oil. Forced lubrication is applied to the top and bottom of the main turbine bearings and to both the forward and after faces of the thrust horseshoes. Oil from the main bearings is caught in oil wipers, and that from the thrust collects in a reservoir in the thrust pedestal, so that the bottoms of the collars run continuously immersed. In the drains from the oil wipers and in the overflow from the thrust-pedestal reservoir are sight glasses through which the flow of oil may be observed. Thermometers are fitted to these sight glasses to show the temperature of the oil leaving the bearings. A cooling coil with water-service connection is provided in the bottom of the thrust-pedestal reservoir.

WATER SERVICE.

Water service is supplied to the engine room by a 3-inch connection from the discharge side of the circulating pump and also by a 1½-inch connection from the fire main. There is

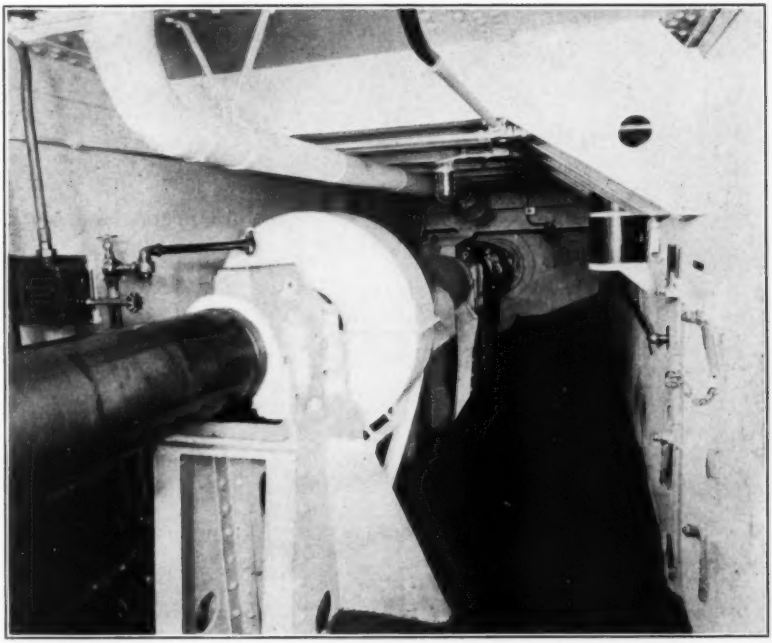


Fig. 6.—FÖTTINGER TORSION METER. "NORTH DAKOTA."

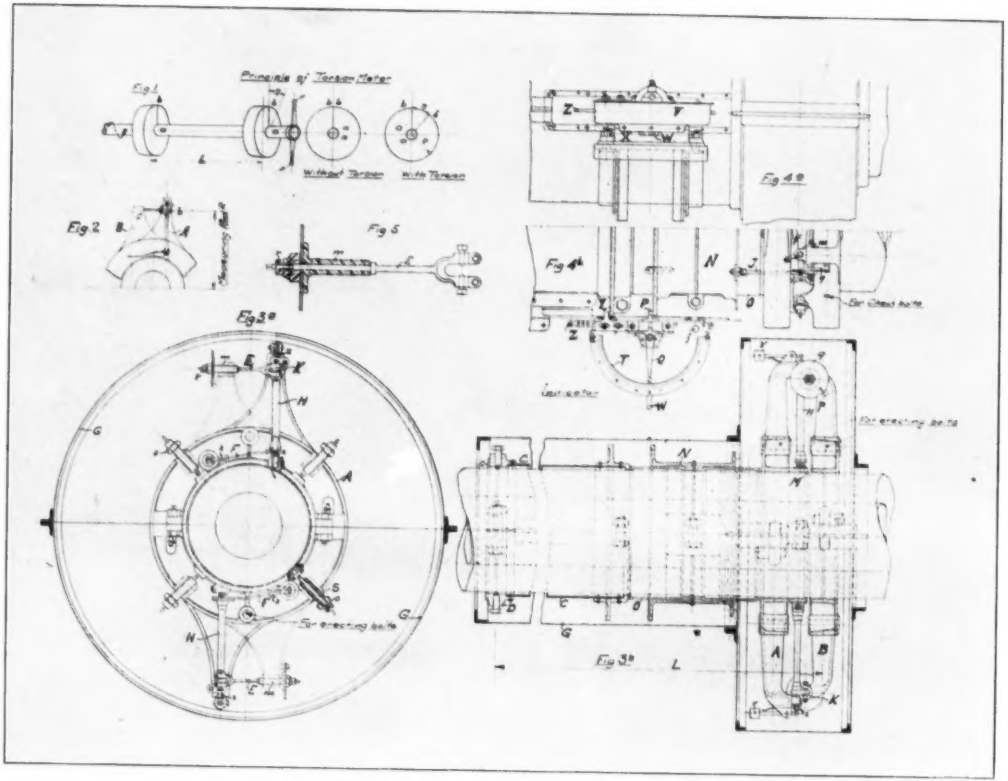


Fig. 7.—DIAGRAM. FÖTTINGER TORSION METER.

a connection to the stern tube, hose connections are supplied in suitable places, flexible elbow connections are fitted to the spring bearings and main turbine bearings, and closed systems are provided for the main turbine bearings, thrust horseshoes, oil reservoir in thrust pedestal, and the cooling coils in oil-drain tanks. Drains from the closed systems lead to a 3-inch pipe connected with the suction of the independent bilge pump.

TORSION METERS.

A Föttinger torsion meter is installed on each intermediate shaft for determining the shaft horsepower of the main turbines by measuring the torsional deflection of the shaft while in motion. A photograph of the installation is shown in Fig. 6, and a sectional drawing of the working parts in Fig. 7. The ring B with two arms and the ring D at the after end of the steel sleeve C are both clamped firmly to the shaft. On the forward end of the sleeve is a ring A similar to B. There is a clearance between this ring and the shaft, and it is centered on B by means of radial centering rods having knife edges on each end. The sleeve C and ring A act as a rigid mass and turn with that part of the shaft to which D is clamped. When a torsional stress is applied to the shaft the ring B twists relative to A. Through a flexible link E, a bell crank H pivoted on A, and a link J connected to the magnalium ring N which floats on the sleeve C, this relative twist is magnified and changed to a fore-and-aft motion. A shoe P is brought into contact with a flange on the magnalium ring, and the fore-and-aft motion of the ring is again magnified and indicated by a pointer Q on a scale T.

The constant for the machine is found by calibrating the shaft and torsion meter in the shop previous to installation in the ship. The after end of the shaft is bolted to a fixed flange. The forward end rests in a roller bearing and on the forward flange is bolted a double lever. A known twisting moment is applied to this lever and the corresponding torsion meter readings are observed. From the slope of the line

Fig. 7.—Diagram. Föttinger Torsion Meter.

plotted between torsion-meter readings and torque a constant is obtained such that

$$\text{H.P.} = C \times R \times F,$$

in which H.P. = shaft horsepower, R = revolutions of the shaft per minute, and F = torsion-meter reading.

COUNTERS AND GAUGES.

In each engine room is fitted a continuous counter driven from the main turbine shaft, and a revolution indicator having two pointers which show the relative speeds of the shafts.

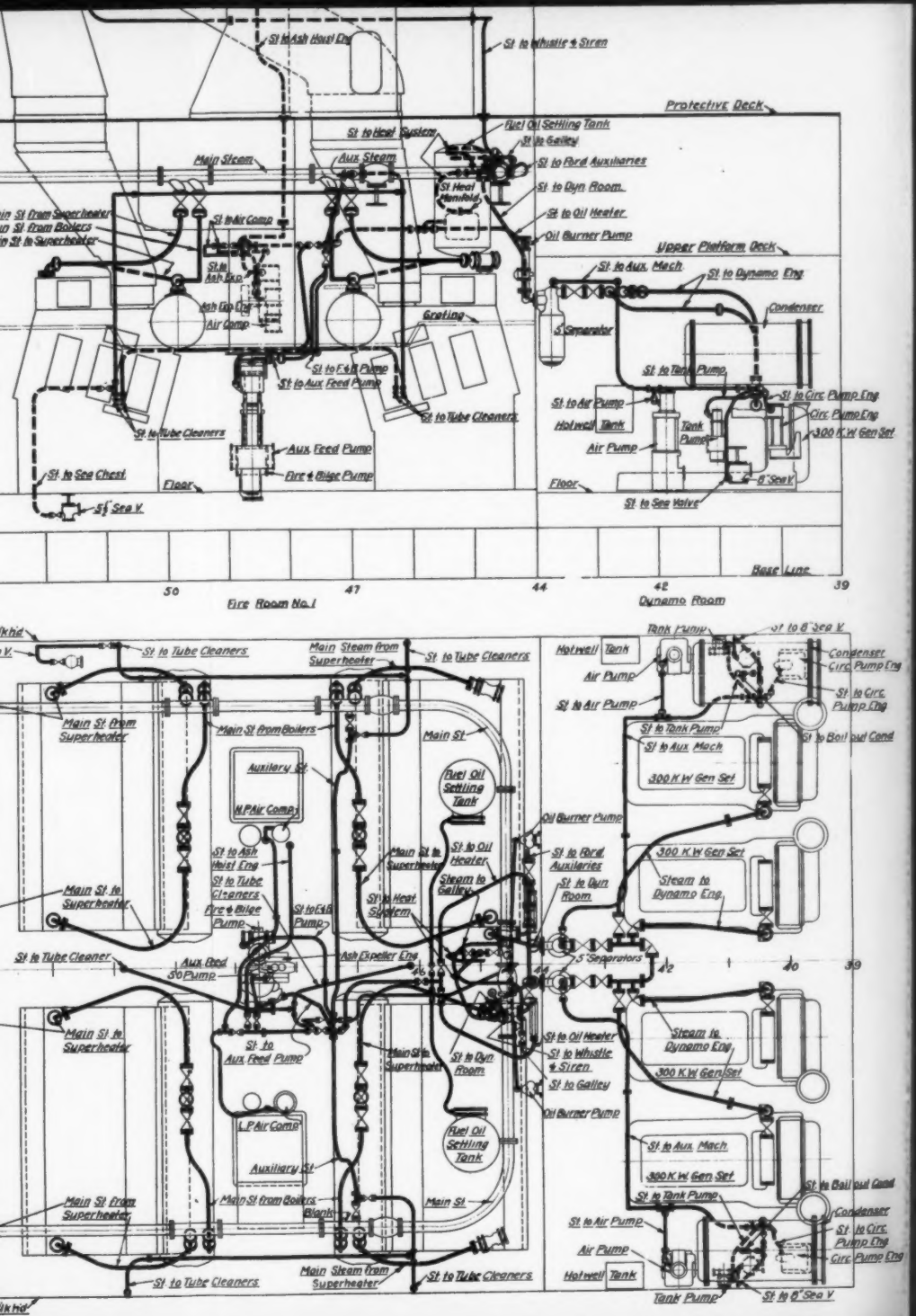
Dial steam gauges are supplied for measuring pressures in the main steam line, turbine steam chest, each turbine stage, and in the condenser. Absolute-vacuum gauges are also connected to the exhaust chamber and the condenser for accurate measurement of vacuum, and a recording steam gauge is connected to the main steam line.

BOILERS.

The boilers are of the well-known Babcock & Wilcox water-tube type, designed and built by that company, and are fitted with superheaters. They are fourteen in number, located in four watertight compartments, two in the after compartment and four in each of the other three. Uptakes from the forward eight lead to the forward stack, and uptakes from the after six to the after stack. The firerooms and boilers are fitted for both coal and oil fuel and the closed-fireroom system is used.

BOILER DATA.

Number of boilers	14
External dimensions, length at base, feet and inches	9–01¼
width at base, feet and inches	16–00¼
height, feet and inches	13–11¼
Number of furnaces, each boiler	1
furnace doors, each boiler	4
headers, each boiler	27
Grate area, each boiler, square feet	102.8
total, square feet	1,439.0


St to Ash Haul Eng
St to Whistle & Siren
Protective Deck
St to Heat System
Fuel Oil Settling Tank
St to Galley
Main Steam
Aux Steam
St to Fwd Auxiliaries
St Heat Manifold
St to Dyn Room
St to Oil Heater
St to Air Comp
Oil Burner Pump
Upper Platform Deck
St to Ash Ejn
St to Aux Mach
St to Dynamo Eng
Ash Ejn Eng
Air Comp
Grating
Condenser
5 Separator
St to F & B Pump
St to Aux Feed Pump
St to Tank Pump
St to Tube Cleaners
St to Air Pump
St to Circ Pump Eng
Hotwell Tank
Circ Pump Eng
Tank Pump
300 K W Gen Set
Aux Feed Pump
Air Pump
St to Sea Chest
Fire & Bilge Pump
6" Sea V
Floor
St to Sea Valve
St to Sea V
Base Line
50
47
44
42
39
Fire Room No 1
Dynamo Room
Main St from Superheater
Main St from Boilers
Main St
Auxiliary St
H P Air Comp
Fuel Oil Settling Tank
St to Ash Haul Eng
Main St to Superheater
St to Oil Heater
St to Tube Cleaners
Steam to Galley
Fire & Bilge Pump
St to F & B Pump
St to Heat System
Ash Ejector Eng
Aux Feed Pump
Steam to Dynamo Eng
Blanks
L P Air Comp
Auxiliary St
Main St from Boilers
Main St
Main Steam from Superheater
St to Tube Cleaners
Tank Pump
Hotwell Tank
Air Pump
St to Air Pump
St to Tank Pump
St to Aux Mach
300 K W Gen Set
St to Boil out Cond
Oil Burner Pump
St to Fwd Auxiliaries
St to Dyn Room
5 Separators
St to Whistle & Siren
St to Galley
Condenser
Circ Pump Eng
St to Circ Pump Eng
St to 6" Sea V

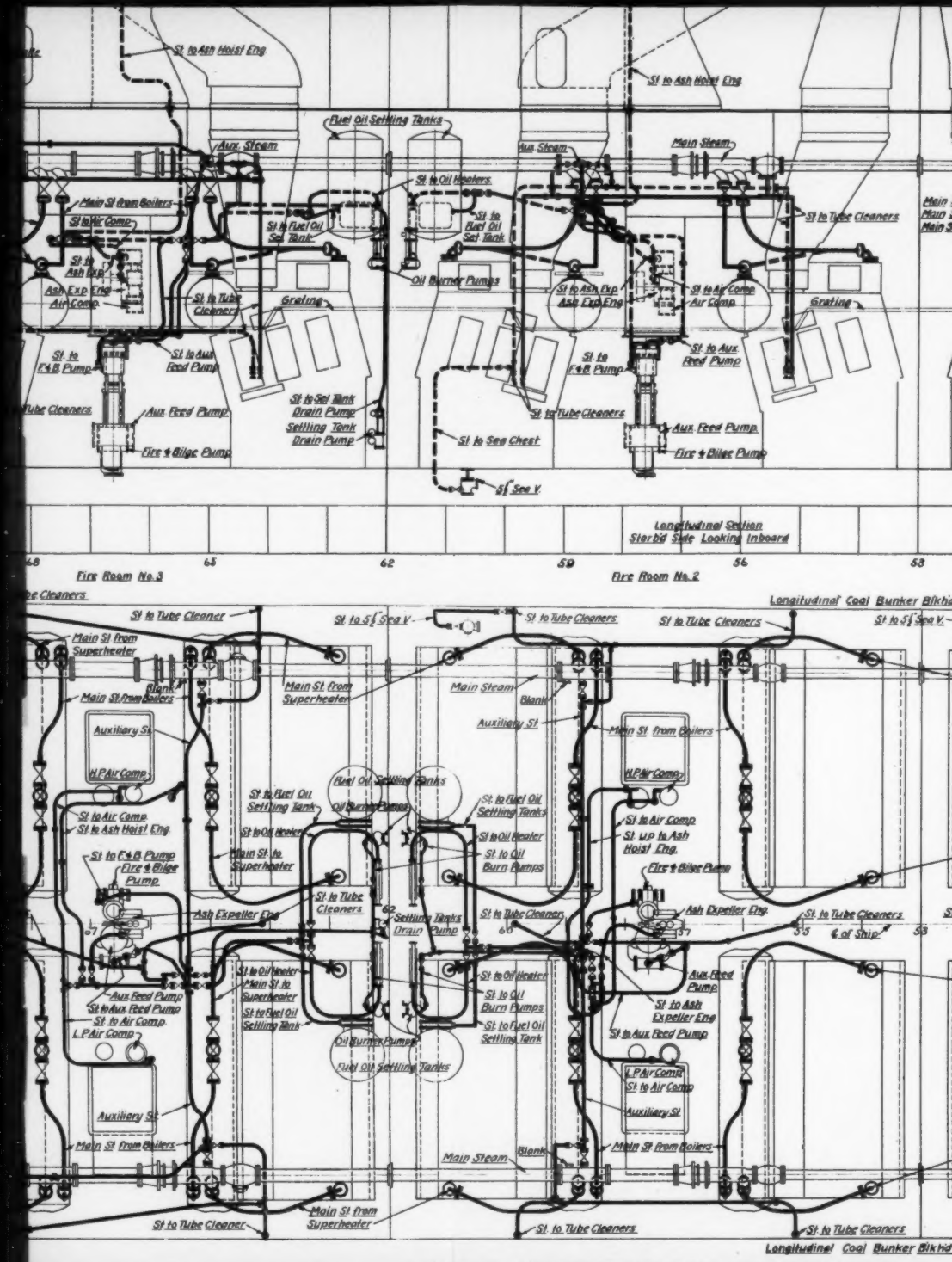


Fig. 8.—Arrangement of Boiler and Dynamo Room.

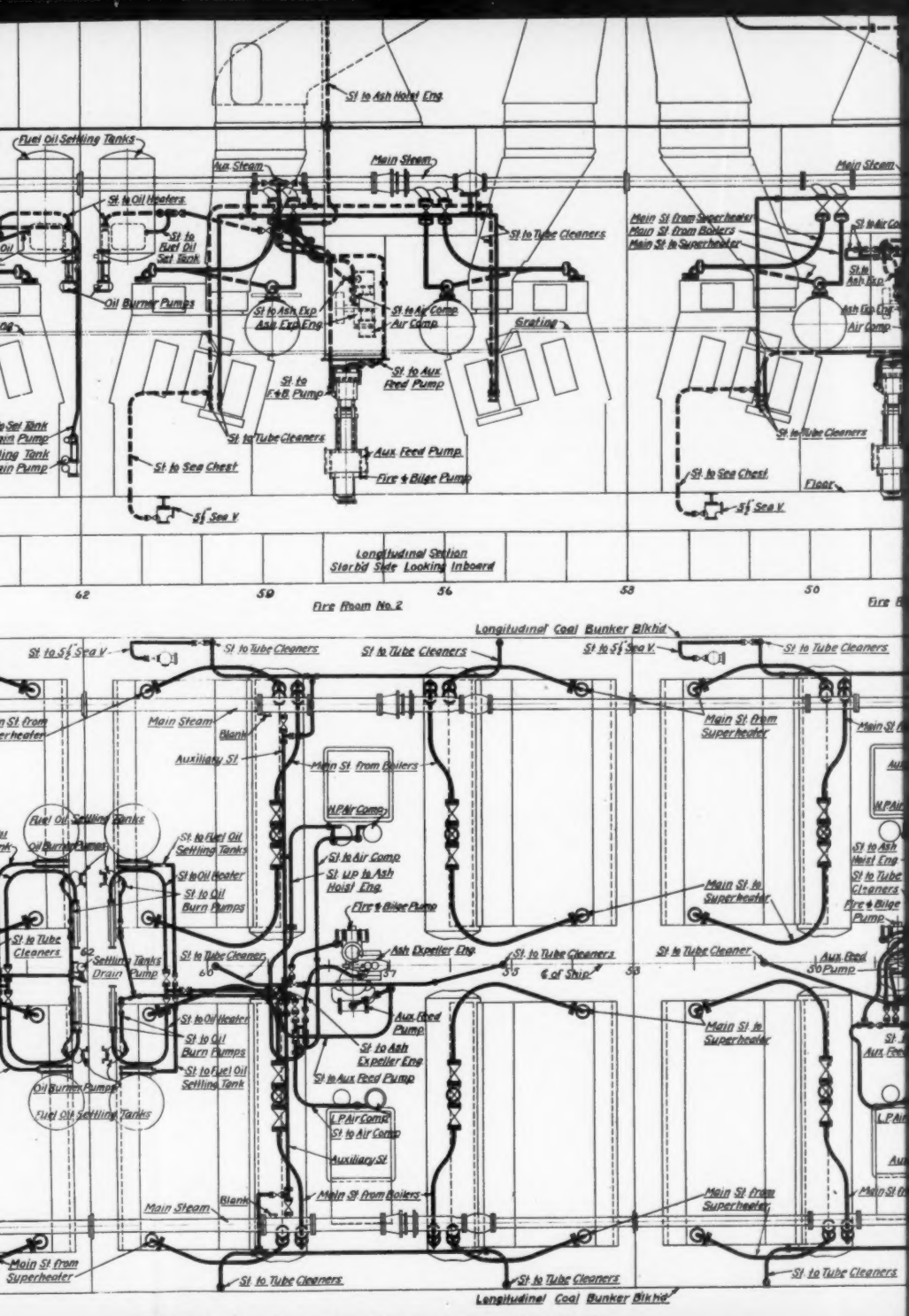
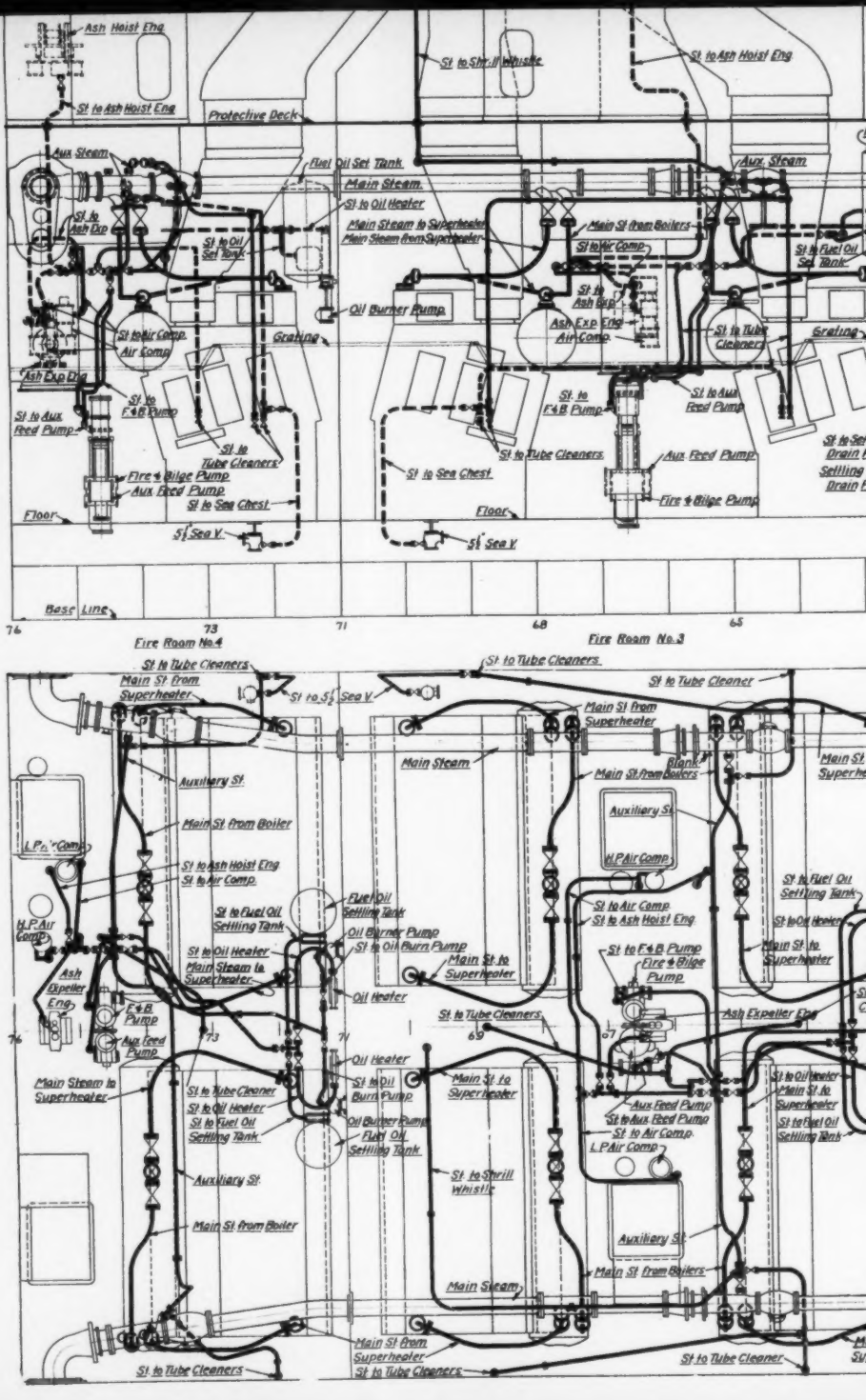


Fig. 8.—Arrangement of Boiler and Dynamo Room.


Ash Hoist Eng
St to Ash Hoist Eng
Protective Deck
Aux Steam
Fuel Oil Set Tank
Main Steam
St to Oil Heater
St to Ash Exp
St to Oil Set Tank
St to Air Comp
Air Comp
Ash Exp Eng
St to F & B Pump
St to Aux Feed Pump
Oil Burner Pump
Grating
St to Tube Cleaners
Fire & Bilge Pump
Aux Feed Pump
St to Sea Chest
Floor
St Sea V
Base Line
St to Shrill Whistle
Main Steam to Superheater
Main Steam from Superheater
Main St from Boilers
St to Air Comp
Ash Exp Eng
Air Comp
St to Fuel Oil Set Tank
St to Tube Cleaners
Aux Feed Pump
Fire & Bilge Pump
Fire Room No 4
Fire Room No 3
St to Tube Cleaners
Main St from Superheater
St to Sea V
Auxiliary St
Main St from Boiler
L P Air Comp
St to Ash Hoist Eng
St to Air Comp
H P Air Comp
Fuel Oil Settling Tank
St to Fuel Oil Settling Tank
St to Oil Heater
Main Steam to Superheater
Oil Burner Pump
St to Oil Burn Pump
Main St to Superheater
Ash Expeller Eng
F & B Pump
Aux Feed Pump
Oil Heater
St to Tube Cleaner
St to Oil Burn Pump
Fuel Oil Settling Tank
St to Shrill Whistle
Main Steam
Blank
Main St from Boilers
Aux Feed Pump
St to Aux Feed Pump
St to Fuel Oil Settling Tank
St to Oil Heater
Main St to Superheater
Ash Expeller Eng
Main St from Superheater
St to Tube Cleaners

Stacks, number	2
area, each, square feet	114.1
height of top above grate, feet and inches	92-01¾
Tubes, number in each boiler, 2-inch (8 B.W.G.)	795
4-inch (6 B.W.G.)	29
Boiler heating surface, total, square feet	55,800.0
Superheater heating surface, total, square feet	6,320.0
Total heating surface, square feet	62,120.0
Boiler heating surface per square foot of grate	38.8
Water-test pressure, pounds gauge	450.0
Working pressure, pounds gauge	295.0

FORCED-DRAFT BLOWERS.

Air is supplied to each of the forward firerooms by four forced-draft blowers and to the after one by two. The blower rooms are airtight compartments, located over the firerooms just beneath the protective deck. Air is taken into each room through wire-mesh panels from the ventilator trunks, the bottom ends of which are closed when the firerooms are under forced draft, and is discharged directly into the firerooms through the blower-room floor.

The blowers are of the Sirocco type with 27-inch-diameter wheels, two 24-inch-diameter inlets, and a 24 × 36-inch outlet opening. They are direct connected to 30-H.P. General Electric motors.

FUEL OIL SYSTEM.

In addition to the ordinary arrangements for burning coal the ship is fitted with a complete oil-fuel system. From storage tanks in the inner bottom the oil is pumped by pumps in the engine rooms to settling tanks in the firerooms. From these tanks it is pumped under pressure through heaters to the burners.

Each oil-storage compartment is supplied with an expansion trunk. The capacities of the various tanks are as follows:

Compartment.	Capacity, gals.	Compartment.	Capacity, gals.
C–92	14,556	C–96	13,935
C–93	14,556	C–97	13,935
C–94	8,490	C–98	15,768
C–95	8,490	C–99	15,768

There is one oil-fuel pump in each engine room for pumping oil to the settling tanks. Both suction and discharge are supplied with Macomb strainers. There are two settling tanks in each fireroom just beneath the protective deck, those in the after room being 33 inches in diameter by 78 inches high over all, and those in the other rooms 43 inches in diameter by 88¼ inches high. Each tank is supplied with steam-heating coils, gauge glasses and drain. The drains of all eight tanks lead to the suction of a common drain pump located on the forward bulkhead in fireroom No. 3, and discharging overboard. There are two oil-burner pumps in each fireroom, which take oil from the settling tanks through Macomb strainers and deliver it to the burners through the heaters. The heaters are of the Schutte & Koerting type, with three tubes. Each boiler is furnished with six Schutte & Koerting burners arranged in three groups of two each between the furnace doors. Provision is made for pumping the cold oil out of the burner pipes and through the heaters before starting the burners.

ASH HOISTS.

One ventilator trunk in each fireroom is fitted with apparatus for hoisting ashes to the main deck. The buckets are lifted by ropes running over sheaves and are guided by vertical guides inside the trunks. Trolleys are provided on the main deck for carrying the buckets after they leave the hoists to the ash chutes on either side of the ship.

The ash-hoist engines are located on the outside of the ventilator trunks and are of the Williamson Bros. two-cylinder reversible type, of sufficient power to hoist 300 pounds from the fireroom floor to the upper deck in 5 seconds with 150 pounds steam pressure. They are controlled from the main deck, and each has an adjustable safety gear to prevent overwinding and to stop the engine when the bucket reaches the fireroom below.

DATA FOR ONE ENGINE.

Number of cylinders	2
Diameter of cylinders, inches	4½
Stroke, inches	4½
Diameter of drum, inches	5
Number of ash-hoist engines in ship	4

ASH-EXPELLER SYSTEM.

In addition to the above described ash hoists there is installed in each fireroom one Stone's patent underline pneumatic ash and clinker expeller. A 4 × 3-inch double steam engine, located overhead of the firing space and running at a normal speed of 200 revolutions per minute, drives by means of gears the combined crusher and expeller located on the fireroom floor. The ashes and clinkers are thrown into a hopper and drop down into the crusher, in which they are broken up by being crushed between a fixed jaw and a movable jaw operated by a cam on the main shaft. Below the crusher is the expeller, which consists of a hollow tapered drum revolving inside of a correspondingly tapered cylinder. The drum is driven by a chain drive from the main shaft and turns at a normal speed of 17 revolutions per minute. When a port in this drum comes opposite a corresponding port in the top of the cylinder a charge of ashes drops down into the drum. When this port has closed compressed air enters the drum and forces the ashes out through the ash port in the drum and a corresponding port in the bottom of the cylinder. An exhaust port is provided to allow the escape of any air that may be caught in the drum before the ash port comes around to receive another charge. This exhaust air escapes to a galvanized-steel tank of about three-quarters of a cubic foot capacity, with a perforated bottom and a door for cleaning. The drum and cylinder are so made that the system is watertight, and the joint between the drum and cylinder is made to hold air pressure. A watertight cover is fitted to the hopper. After the ashes leave the drum they are forced out clear of the skin of the ship through a gate valve and a sea chest. The gate valve is kept closed when the expeller is not in use.

Each expeller has its own air-compressor plant, consisting of a low-pressure compressor, a high-pressure compressor, an intermediate reservoir and a main reservoir. The low-pressure compressor, $9\frac{1}{2} \times 13 \times 10$ inches, discharges to the intermediate reservoir through about 25 feet of $1\frac{1}{4}$-inch cooling pipe. The high-pressure compressor, $9\frac{1}{2} \times 9 \times 10$ inches, takes air from the intermediate reservoir and discharges through a similar cooling pipe to the main reservoir. The intermediate reservoir is 16 inches diamter by 42 inches high, and the main reservoir is 12 inches diameter by 60 inches high. Each is supplied with a gauge and drain, and the pressure in the main reservoir is controlled by an automatic governor on the steam line to the high-pressure compressor.

ELECTRICAL FIRING DEVICE.

To aid the systematic firing of the boilers an electric timing device is installed. By means of a motor-driven transmitter in the engine room, which can be regulated for any desired period of firing, gongs are sounded and numbers flashed on indicators conveniently placed in the firerooms to show when each furnace is to be fired.

MAIN STEAM LINE.

The main steam line runs across the forward end of the forward fireroom and aft through the other three firerooms along each outboard bulkhead, gradually increasing in size from 8 inches in the forward room to $13\frac{1}{2}$ inches, where it leaves the after room. On top of each boiler drum is a $5\frac{1}{2}$-inch boiler-stop valve provided with an operating gear controlled from the protective deck, and the arrangement of piping is such that steam may flow from the boiler drum to the main steam line directly or through the superheater. After the main steam line leaves the after fireroom it passes outboard and around a powder magazine through a special steam-pipe passage into the forward end of the engine room through the outboard bulkhead. Steam-pipe insulators cooled by water

service are used in the bulkheads wherever it is necessary to prevent heating. After the steam enters the engine room it passes successively through a separator, stop-valve, strainer, butterfly valve and throttle valve to the main turbine. A 6½-inch cross-connection joins the port and starboard separators, and from the cross-connection pipe 4½-inch steam bleeders lead aft to the main-turbine exhaust pipes. The main steam pipes and fittings are of steel throughout.

AUXILIARY STEAM LINE.

In each fireroom a 2½-inch steel auxiliary steam pipe runs from a fitting in the main steam line on one side directly across the fireroom to a similar fitting on the opposite side. In each engine room a 7-inch steel auxiliary steam line leaves the main steam line at the separator and passes aft and across the after end of the engine room to the center-line bulkhead, where there is a cross connection to the corresponding line in the other engine room. From this line there is a steam connection to the after auxiliaries.

AUXILIARY EXHAUST LINE.

A 7-inch copper auxiliary-exhaust steam line runs across the forward end of the forward fireroom and aft along both port and starboard bulkheads of the other rooms. From the after fireroom it passes outboard around the magazine through the steam-pipe passage, enters the engine room near the main steam pipe, passes aft and across the after end of the engine room to the center-line bulkhead, where there is a cross connection to the other engine room. Connections are provided to direct the exhaust steam into either main condenser, either auxiliary condenser, either feed heater or into the atmosphere.

HEATING SYSTEM.

The heating system consists of nine circuits in two groups, each group having a separate supply manifold and a separate drain manifold. The supply manifold for circuits 1 to 6, in-

clusive, is in the forward end of the forward fireroom, and the manifold for 7 to 9, inclusive, is in the after inboard corner of the port engine room. In each case there is a stop valve and a reducing valve between the auxiliary steam line and the manifold. The reducing valves reduce the pressure to about 50 pounds.

The plumbing circuits drain into the auxiliary-exhaust line through stop valves, and the radiator circuits to the drain manifolds through check valves and stop valves. Each drain mainifold has an independent trap which drains through the trap-drain main to the feed tank, the auxiliary condenser or the main condenser. All steam-heating pipes are brass, iron-pipe size, with screwed fittings. Copper U bends are provided wherever necessary to permit of free expansion.

The radiators are all brass pipe, most of them 1 inch, a few of them 2-inch. Those of 1-inch pipe are made by bending the pipe with a special bending machine instead of using standard screwed return bends which are used on the 2-inch type. Wherever more than 10 square feet of heating surface are required a double coil is used, joined by fittings provided with unions to allow of easy separation of the coils. Each radiator is supplied with a $\frac{1}{2}$-inch steam valve, a $\frac{1}{2}$-inch drain valve and an air cock. The valves are operated by a socket wrench in the crew space and by hand wheels in the officers' quarters.

REFRIGERATING PLANT FOR COLD STORAGE.

The cold-storage rooms are cooled by two horizontal Allen dense-air ice machines, each with a cooling capacity of two tons of ice per day. Cooling air is supplied to an ice-making tank, scuttle butts, a vestibule with distributing manifold, a meat room, fish room, vegetable room and a room for butter and eggs. Pipes and valves are provided so that the air may go to the cold-storage rooms direct, or through the ice-making tank and thence to the cold-storage rooms and scuttle butts, or from the ice-making tank directly to the scuttle butts. The pipes leading from the ice machines to the tank and cold-

storage rooms are of copper, and the cooling pipes in the rooms are 1-inch galvanized wrought iron. The areas of interior surfaces and cooling surfaces are as follows:

Meat room, interior surface, square feet	530
cooling surface, square feet	321
Fish room, interior surface, square feet	233
cooling surface, square feet	136
Vegetable room, interior surface, square feet	306
cooling surface, square feet	131
Room for butter and eggs, interior surface, square feet	204
cooling surface, square feet	88

EVAPORATING AND DISTILLING PLANT.

The evaporating plant is located on the protective deck above the forward end of the engine-room space and the distillers are placed in the engine hatch at the height of the gun deck. There are four evaporators with a combined capacity of 25,000 gallons of water per 24 hours and four distillers with a combined capacity of 25,000 gallons of potable water per 24 hours.

Each evaporator has 100 brass tubes, 2 inches outside diameter, No. 12 B.W.G., 73⅞ inches long, with a total heating surface of 312 square feet. They are secured to the tube sheets in such a manner that provision is made for expansion. Trolleys are provided for the ready removal of the tubes, and their arrangement is such that their removal will leave the shell accessible in all its parts for scaling. Each evaporator is supplied with a blow pipe for blowing the brine overboard.

Each distiller has 175 brass tubes, ⅝-inch outside diameter, No. 16 B.W.G., 4 feet 5⅞ inches long with a total cooling surface of 125 square feet. They are tinned on both sides. A flanged tube sheet at one end working in a stuffing box allows free expansion of the tubes.

Steam is supplied to the evaporators from the auxiliary steam line through a reducing valve at a working pressure of 80 pounds, and drains through traps to the feed tanks, each evaporator having an independent trap. The evaporator-feed pump takes water from the distiller circulating-pump suction

from the sea and discharges through a 1½-inch feed pipe into the side of the evaporator shell. Vapor leaves the top of the shell and enters the auxiliary-exhaust line or passes through separators to the distillers. The separators and distillers drain to a 100-gallon reservoir tank. Connections are provided for operating the distillers either in parallel or in series. The distiller fresh-water pump takes water direct from the distiller or from the reservoir and discharges to the feed tanks or to the fresh-water tanks. Water is pumped direct from the sea through the distillers by the distiller circulators and is discharged overboard. There is also a connection from the distiller circulating-pump discharge to the flushing system. Water from the engine room fire and bilge pumps may be circulated through the distillers.

AIR-COMPRESSOR PLANT FOR PNEUMATIC TOOLS.

On the center-line bulkhead, above the grating in each engine room, are three 11 × 11 × 12-inch Westinghouse steam-driven air compressors. The six compressors have a combined capacity of 250 cubic feet of free air per minute at 125 pounds pressure. A working pressure of 100 pounds is maintained by an automatic governor on each compressor. The three compressors in each engine room discharge in parallel into two air reservoirs, 24½ inches in diameter by 78 inches high, with a capacity of about 34,000 cubic inches each. These reservoirs supply air to the following systems:

A line fore and aft along the center-line bulkhead below the grating in the port engine room, branches from which lead to valves and hose connections in both engine rooms for pneumatic tools.

A line running to the workshop, with suitable hose connections.

A ½-inch line to firerooms, for blowing soot off the boiler tubes.

A hull gas-ejector system, for ejecting smoke from the guns.

MACHINE SHOP.

The machine shop is located on the protective deck in a separate compartment between the engine-room hatches. The following machines are installed:

One 24 × 46-inch × 11-foot extension gap lathe.
One 14-inch × 8-foot screw-cutting back-geared tool-room engine lathe.
One 15-inch universal geared column tool-room shaper.
One 28-inch upright drill.
One 16-inch sensitive drill.
One universal milling machine.
One 12-inch double emery grinder.
One portable cylinder-boring machine.
Six machinist's vises.

Each machine is driven by an independent enclosed electric motor.

In addition to the above machine tools are furnished a steel blacksmith's forge of the portable folding type with all necessary tools and fixtures and a 140-pound anvil.

ELECTRIC PLANT.

The dynamo room is located forward of the firerooms, directly below the switchboard room and central station, all being under the protective deck. The generator plant consists of four 300-kw. turbo-generators built by the General Electric Company. They are of the eight-pole compound-wound type, each capable of delivering at full load 2,400 ampères of current at 125 volts with a speed of 1,500 revolutions per minute. They are driven by two-stage Curtis turbines with three rows of moving buckets in each stage. The valve mechanism for controlling the speed is of the crosshead mechanical type, consisting of six valve units operated by an oscillating lever driven from the main turbine shaft through a double reduction-worm gear, and controlled by an inertia governor mounted on the primary gear shaft. An emergency governor is also furnished which closes the throttle valve in case of failure of the

regular governing device to prevent an increase of speed of more than 8 per cent. above the normal.

Steam is supplied to the turbines by two 5-inch pipes from the main steam line in the forward end of the forward fireroom, one supplying the two port and the other the two starboard turbines. There is a separator in each line and a cross connection between the two lines. There is one condenser on each side of the dynamo room. In addition to a separate exhaust pipe from the exhaust chamber of each turbine to its condenser there is also a 10-inch pipe leading to a common atmospheric exhaust. An independent air pump takes water from the condenser and discharges it to a hotwell tank, from which it is pumped to the main feed tank in the engine room. The circulating pump is of the centrifugal type driven by a 5×5-inch vertical steam engine.

The dynamo condenser is of the return-flow type with a shell of $\frac{3}{8}$-inch steel plate 46 inches inside diameter, having a rectangular exhaust nozzle 20×54 inches. The heads are of composition, the forward one being divided into an upper and lower chest with 7-inch injection and discharge nozzles. The tube sheets are of composition with glands similar to those in the main condenser. Each condenser has 1,574 composition tubes, $\frac{5}{8}$-inch outside diameter, No. 16 B.W.G., 7 feet 2 inches long, with a total cooling surface of 1,800 square feet.

ANCHOR WINDLASS.

The anchor windlass is of the vertical type, having two vertical shafts driven by a worm gearing direct from a worm on the crank shaft of the engine. Each shaft has on its upper end, above the forecastle deck, a wildcat, and the arrangement is such that the wildcats can be operated together or independently of each other. The windlass engine is a double 17×14-inch vertical engine. The working steam pressure is 150 pounds, but the engine is designed to withstand full boiler pressure.

STEERING GEAR.

The main steering gear consists of a right-and-left-handed screw on which are two driving nuts direct connected by side rods to the crosshead on the rudder stock. The screw is coupled by a lug clutch to another section of shafting driven through a pair of herring-bone gears by a crank shaft of a horizontal double 17×14-inch engine. The engine is designed for 150 pounds working steam pressure, but is capable of withstanding full steam pressure. For power steering the valves of the steering engine are controlled from the steering stands in the conning tower, on the upper conning platform, and in the central station through wire rope transmission, and also from the central station by telemotor. The valves may also be operated by a handwheel in the steering-engine room. Provision is made for hand steering by four large handwheels forward of the steering engine on a shaft geared to the screw shaft. An arrangement of relieving tackles direct connected to the crosshead is also operated by these handwheels.

OFFICIAL TRIALS—STANDARDIZATION.

The ship was standardized on the measured-mile course off Rockland, Maine, November 4th, 1909.

Draught at beginning of trial, forward, feet and inches	26–11⅞
aft, feet and inches	27–01¼
mean, feet and inches	27–00$\frac{5}{16}$
end of trial, forward, feet and inches	26–06
aft, feet and inches	27–00⅜
mean, feet and inches	26–09$\frac{11}{16}$
Displacement at beginning of trial, tons	20,126
end of trial, tons	19,924
Estimated displacement at middle of five high-speed runs, tons	20,051
Corresponding draught, feet and inches	26–11⅛
Trim, inches by stern	3

Data for standardization runs is recorded in Table II and plotted on accompanying curve, Sheet I.

For a period of two hours, including the high-speed runs, oil fuel was used in combination with coal.

POWER OF AUXILIARIES.

Complete tests were made on the auxiliaries previous to the trial, the results of which are shown on the accompanying curve sheets 2–12.

The I.H.P. chargeable to the motor-driven forced-draft blowers is found by adding to the power required to drive them as indicated by the switchboard a proportionate amount of I.H.P. for auxilaries in the dynamo room. The steam consumption chargeable to the blowers is obtained by multiplying this value for I.H.P. by the total water rate of the generator plant corresponding to total output.

Table I.

TABLE I. PUMPS. U.S.S. NORTH DAKOTA.

Auxiliary Pumps	Number	Type and Make	Size	Location
Main Air, Wet	4	Weir Manotype, Vertical, Piston, Single acting.	15 x 30 x 16	Two in each Engine Room.
Main Air, Dry.	2	Weir Twin, Vertical, Rotative, Single acting.	36 x 20 x 7	One in each Engine Room.
Main Circulating	2	Centrifugal	48 inch runner	One in each Engine Room.
Main Feed	4	Fore River, Vertical, Piston, Double acting, Simplex.	16 x 9½ x 18	Two in each Engine Room.
Auxiliary Feed	3	Fore River, Vertical, Piston, Double acting, Simplex.	16 x 9½ x 18	One in each of three Port Firerooms.
	1	Blake, Vertical, Piston, Double acting, Simplex.	10 x 7 x 12	One in after Fireroom.
Fire and Bilge	8	Blake, Vertical, Piston, Double acting, Simplex.	12 x 10 x 18	One in each Fireroom. Two in each Engine Room.
Independent Bilge	2	Blake, Vertical, Piston, Double acting, Simplex.	6 x 8 x 12	One in each Engine Room.
Auxiliary Condenser	2	Blake, Horizontal, Simplex, Combined Air and Circulating	8 x 22 x 10 x 12	One in each Engine Room.
Forced Lubrication	4	Blake Vertical, Piston, Double acting, Simplex.	6 x 7 x 8	Two in each Engine Room.
Insulator Circulating Water	2	Blake Vertical Piston, Double acting, Simplex.	6 x 8 x 8	One in each Engine Room.
Engine Room Fuel Oil	2	Blake, Vertical Piston, Double acting, Simplex.	6 x 7 x 8	One in each Engine Room.
Oil Burner	8	Blake, Vertical, Piston, Double acting, Duplex.	4½ x 12½ x 4	Two in each Fireroom.
Oil Settling Tank Drain	1	Fore River, Vertical, Piston, Double acting, Simplex.	9½ x 4 x 4	One in No. 3 Fireroom.
Dynamo Condenser Circulating	2	Centrifugal	36 inch runner	Two in Dynamo Room.
Dynamo Air	2	Blake, Featherweight, Vertical, Piston, Single acting, Simplex.	8 x 16 x 12	Two in Dynamo Room.
Dynamo Hotwell	2	Blake, Vertical, Piston, Double acting, Simplex.	4½ x 5 x 6	Two in Dynamo Room.
Evaporator Feed	1	Blake, Vertical, Piston, Double acting, Simplex.	4½ x 3 x 6	One in Evaporator Room, Port.
Distiller Fresh Water	1	Blake, Vertical, Piston, Double acting, Simplex.	4½ x 5 x 6	One in Evaporator Room, Stbd.
Distiller Circulating	2	Blake, Horizontal, Piston, Double acting, Simplex.	12 x 14 x 12	Two in Evaporator Room.

On all trials water was pumped from each feed tank to measuring tanks in the engine hatch and drained to a watertight compartment outboard from each engine room.

The coal was weighed and bagged previous to the trial, each bag holding 112 pounds.

U.S.S. NO

CURVES FOR OFFIC

ROCKLAND, M

MEAN DISPLACEME

—PROP

DIAMETER----------

PITCH----------

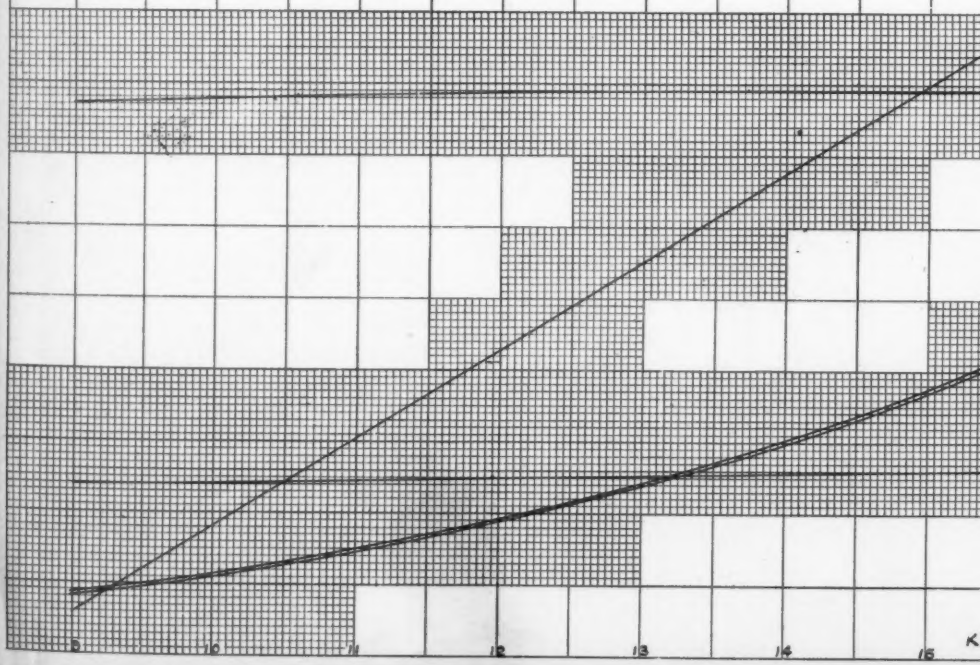
PITCH RATIO----------

DEVELOPED AREA, SQ

PROJECTED AREA, SQ

DEVELOPED AREA ÷ D

NO. OF BLADES----

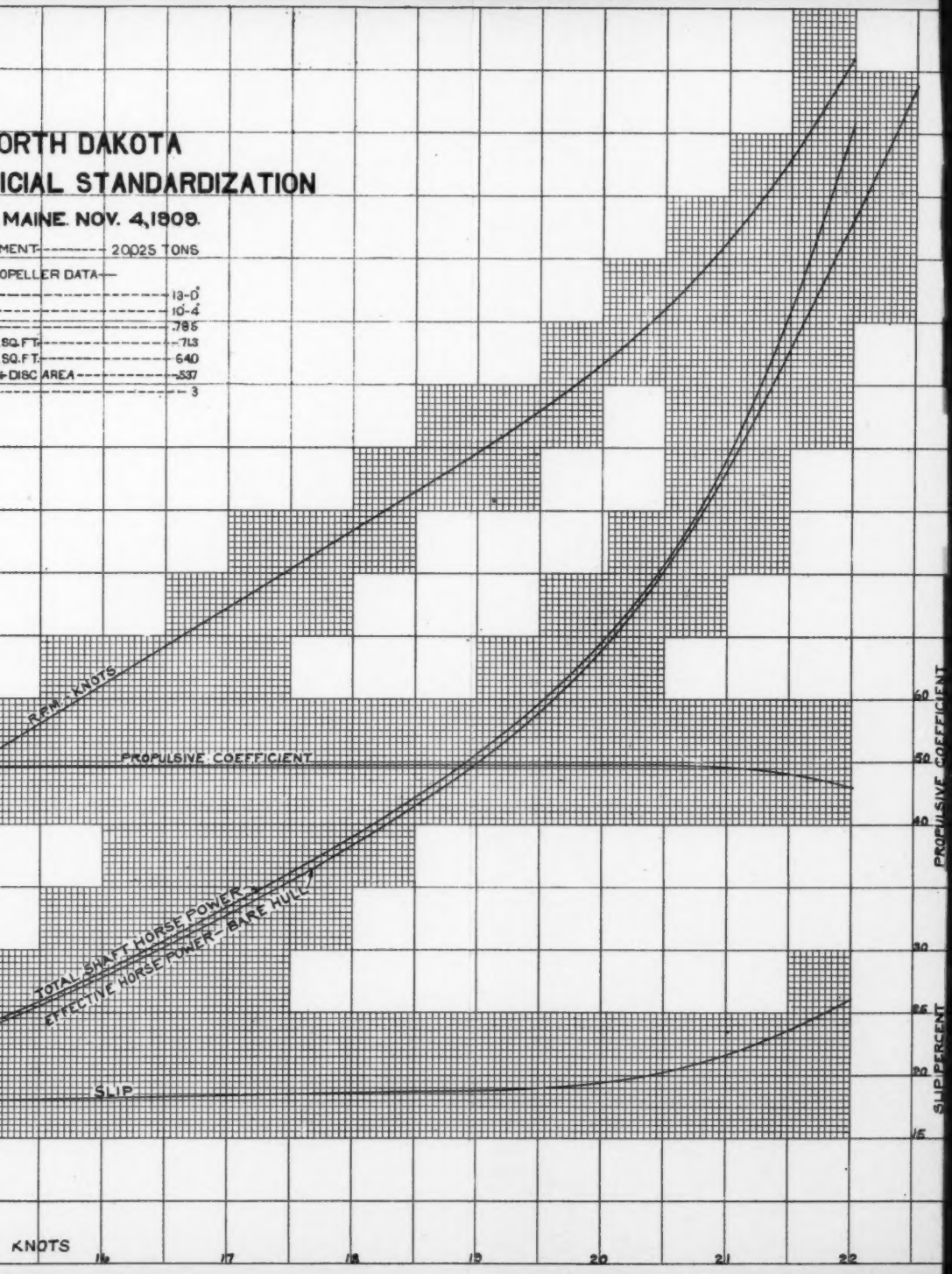


ORTH DAKOTA
ICIAL STANDARDIZATION
MAINE. NOV. 4, 1908.

MENT-------- 20025 TONS

OPELLER DATA—

----------	13-0
----------	10-4
----------	.785
SQ.FT. ----------	.713
SQ.FT. ----------	640
+ DISC AREA ----------	.537
----------	3



Sheet 1

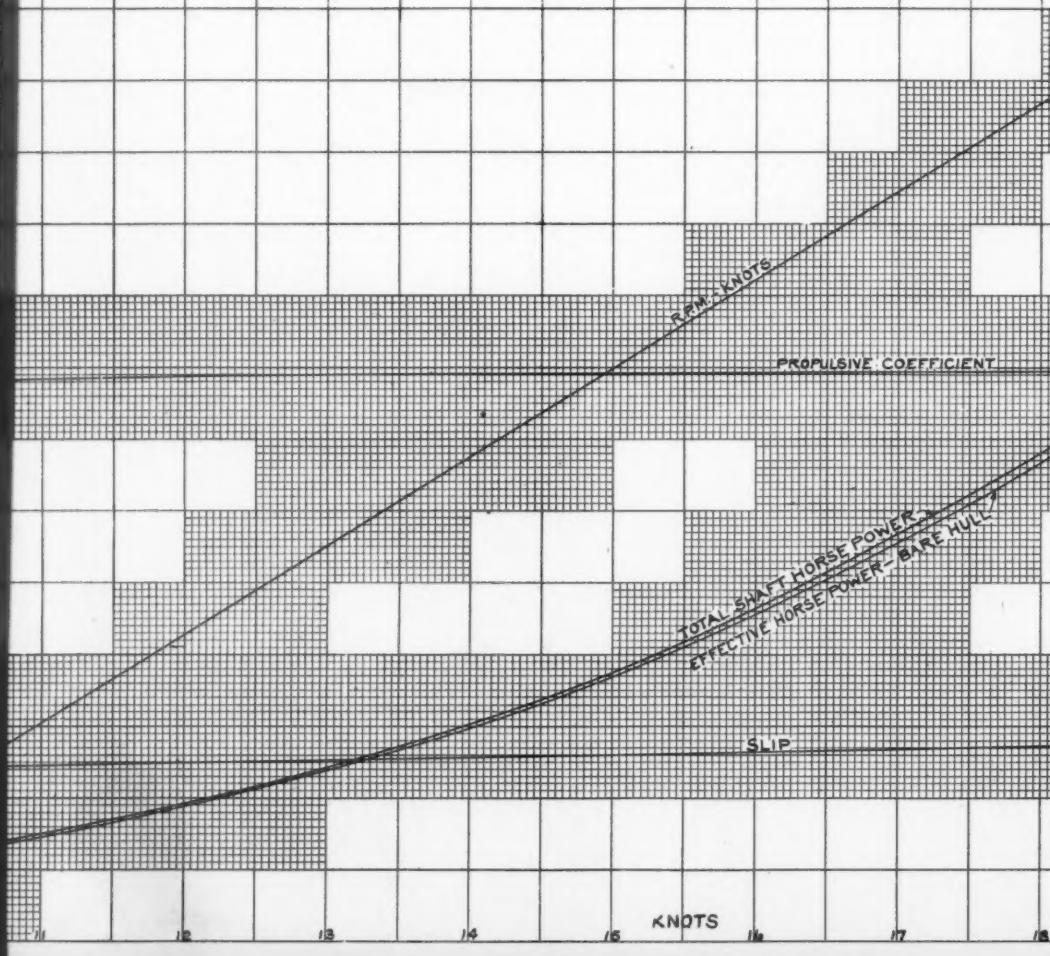
U.S.S. NORTH DAKOTA
CURVES FOR OFFICIAL STANDARDIZATION

ROCKLAND, MAINE. NOV. 4, 1908.

MEAN DISPLACEMENT ---------- 20025 TONS

—PROPELLER DATA—

DIAMETER	18-0"
PITCH	10-4"
PITCH RATIO	.785
DEVELOPED AREA, SQ. FT.	713
PROJECTED AREA, SQ. FT.	640
DEVELOPED AREA ÷ DISC AREA	.537
NO. OF BLADES	3



Sheet 1.

SLIP PERCENT
PROPULSIVE COEFFICIENT
EFFECTIVE HORSE POWER
SHAFT HORSE POWER
REVOLUTIONS PER MINUTE
C.S. 1151
18
19
20
21
22

TABLE II U.S.S. NORTH DAKOTA

Data from progressive runs over measured mile course off Rockland, Maine. November 4, 1909.

				Starboard Turbine							Port Turbine							Both Turbines.		Averages by Groups		
Group No.	Run No.	Direction	Speed Knots.	Nozzles open.	Chest Press. Abs.	Chest Temp.	Exhaust Press. Inches	R.P.M.	Torsion Meter.	B.H.P.	Nozzles open.	Chest Press. Abs.	Chest Temp.	Exhaust Press. Inches.	R.P.M.	Torsion Meter.	B.H.P.	Mean R.P.M.	Total B.H.P.	Knots.	R.P.M.	B.H.P.
	1	S	22.032	20	267.6	453.	26.60	278.47	152.33	15879	20	264.6	435	264	280.02	126.75	13167	279.74	29046			
	2	N	20.761	20	238.1	458.	27.12	271.28	142.22	14391	20	234.6	457	268	273.70	137.00	13909	272.49	28300			
	3	S	22.222	20	268.6	469.	26.80	286.29	157.16	16782	20	266.6	445	260	294.44	138.93	14661	285.36	31443			
	4	N	21.404	20	273.6	466.	26.73	287.19	158.75	17005	20	266.6	470	265	287.90	153.46	16390	287.54	33395			
1	5	S	22.154	20	264.6	462.	26.85	283.87	154.92	16409	20	260.6	480	264	283.97	149.87	15896	283.97	34305	21.829	286.33	33960
	6	N	21.570	20	272.6	466.	26.70	286.13	157.34	16791	20	264.6	472	263	287.20	149.50	15860	286.66	34851			
	7	S	22.164	20	278.6	458.	26.67	288.40	158.21	17090	20	267.6	478	263	289.51	148.00	15844	288.95	35134			
	8	N	20.571	14	263.6	469.	27.40	252.74	118.14	11137	14	265.6	466	272.5	256.17	122.00	11595	254.45	22732			
2	9	S	20.455	14	260.6	458.	27.56	251.27	117.55	11016	14	254.6	464	272.1	251.62	120.46	11245	251.44	22261	20.636	252.88	22625
	10	N	20.666	14	264.6	459.5	27.40	262.49	118.00	11207	14	264.6	465	272.0	255.34	126.82	12042	254.21	23249			
	11	S	18.780	13	256.6	450.	27.70	243.98	108.44	9868	13	254.6	463	274.5	242.00	107.17	9621	242.99	19489			
3	12	N	19.034	13	236.6	450.5	27.85	237.21	102.57	9072	13	236.6	455	275.0	239.20	107.10	9504	238.20	18576	18.771	238.59	18888
	13	S	19.498	13	236.1	440.	27.75	238.49	102.75	9140	13	239.6	453	274.5	239.47	110.00	9773	238.98	18913			
	14	N	18.246	10	222.6	442.	28.10	213.04	78.50	6238	10	234.6	447	278.5	216.85	83.46	6714	214.94	12952			
4	15	S	16.877	10	201.1	435.	28.03	207.87	77.00	5970	10	215.6	485	279.0	207.31	78.00	6076	207.59	12046	17.346	208.74	12108
	16	N	17.383	10	208.6	431.	28.70	204.57	73.63	5618	10	207.6	490	278.0	205.16	75.84	5772	204.86	11390			
	17	S	11.713	4	213.6	418.5	28.45	139.56	32.36	1684	4	217.6	420	282.0	141.71	35.38	1860	140.63	3544			
5	18	N	11.561	4	208.6	402.	28.42	132.37	31.08	1534	4	215.6	390	282.2	136.03	34.18	1725	134.20	3259	11.432	135.73	3389
	19	S	10.806	4	209.6	404.	28.46	131.90	31.86	1568	4	215.6	415	279.7	136.79	34.29	1727	133.89	3296			
	20	N	10.607	3	222.6	406.	28.41	120.93	25.68	1158	3	224.6	408	281.0	122.00	27.00	1222	121.46	2380			
6	21	S	8.863	3	223.6	408	28.23	118.36	26.92	1188	3	244.6	406	281.1	120.60	27.61	1234	119.43	2422	10.198	120.67	2423
	22	N	10.465	3	223.6	411.5	28.33	121.75	26.82	1218	3	224.6	406	281.2	122.37	27.42	1251	122.36	2469			
	23	S	12.397	4	253.6	428.	28.15	146.48	37.82	2066	4	244.6	401	281.00	146.30	37.65	2043	146.39	4109			
7	24	N	11.380	4	227.6	421.	28.28	141.37	35.40	1866	4	213.6	405	281.00	139.57	34.47	1785	140.47	3651	12.036	141.33	3748
	25	S	11.788	4	221.6	418.	28.05	136.54	34.12	1737	4	234.6	395	281.00	139.48	35.61	1843	138.01	3580			

FOUR-HOUR FULL-POWER TRIAL.

Time of beginning of trial	9:00 A. M., Nov. 5, 1909.
end of trial	12:00 M., Nov. 5, 1909.
Estimated average displacement during trial, tons	20,020
Corresponding mean draught, feet and inches	26-10⅞
Trim, inches by stern	5
Course of ship	S.W.
Wind	Stiff to fresh breezes from N.W.
Sea	Smooth to moderate.
Average barometer reading, inches	29.77

GENERAL NOTES.

Auxilaries in engine room were operated as shown by "Averages of Observations."

The ice machines were not running.

Live steam was used in the port feed heater and drained to the starboard auxiliary condenser.

The auxiliary exhaust led into the starboard feed heater and starboard main condenser, and finally to the starboard feed tank.

The port measuring tanks received water from port turbine only.

The starboard measuring tanks received water from the starboard turbine and from all auxiliaries whose power was required, by contract and trial-trip precept, to be used for determining water-consumption results.

A tube blew out in forward starboard boiler, No. 3 fireroom, about 12:25 P. M., causing the whole fireroom to be put out of commission. The trial was continued to the end of the four hours, but on account of this irregularity the following data is given for the first three hours only of the trial.

Pressures. (*Average of 15-minute observations.*)

	Starboard.	*Port.*
Main steam at boilers, pounds gauge	284.0	
in engine room, pounds gauge	252.7	254.8
turbine steam chest, pounds gauge	248.7	241.3
fifth-stage receiver, pounds absolute	15.7	14.3
exhaust chamber, pounds absolute	1.54	1.60
condenser, pounds absolute	1.22	1.21

	Starboard.	*Port.*
Dry-air-pump steam chest, pounds gauge	47.5	61.2
Main feed line, pounds gauge	332.0	329.0
Auxiliary exhaust in heater, pounds gauge	2.6	...
Forced-lubrication system, pounds gauge	19.8	22.0
Engine-room bilge-pump discharge, pounds gauge	11.0	...
Dynamo-turbine steam chest, pounds gauge	213.0	...
condenser, inches of mercury	28.5	...
Air pressure in firerooms, inches of water	1.9	

Temperatures. (*Average of 15-minute observations.*)

Steam at turbine chest, degrees F	459.6	469.1
superheat, degrees F	54.1	66.1
Injection, degrees F	49.0	49.0
Overboard discharge, degrees F	86.0	86.0
Heater inlet, degrees F	93.7	78.3
outlet, degrees F	154.0	101.0
Oil from main-turbine bearings, mean for two bearings, degrees F	111.5	112.1
Engine-room air, degrees F	94.0	98.0
Firerooms, working level, degrees F	87.0	
Smoke stacks, by pyrometer, degrees F	650.0	

Revolutions, or double strokes, per minute. (*Average of 15-minute observations.*)

Main turbines	280.0	280.8
air pumps, wet, forward	42.1	33.7
aft	35.0	44.4
dry	270.6	243.3
feed pumps, inboard	25.5	38.8
outboard	36.0	32.8
circulating pumps	215.6	195.7
Forced-lubrication pumps	10.2	10.5
Independent bilge pumps	28.9	21.9

All the auxiliaries above recorded were in continuous operation for the first three hours, except the starboard inboard feed pump, which ran one hour, and the port outboard feed pump, which ran two hours.

Miscellaneous. (*Average of 15-minute observations.*)

	Starboard.	*Port.*
Number of first-stage nozzles open in main turbine	19.8	19.8
Torsion-meter reading, in millimeters (average of 3-minute observations)	152.01	...
Torsion-meter constant	0.373	0.371
Number of generators in use	1	...

	Starboard.	*Port.*
Voltage at switchboard	125.0	...
Total output of generator, ampères	2,550.0	...
Power output for forced-draft blowers, ampères	2,125.0	...
Speed of ship, knots per hour	21.64	
Slip of propeller, in per cent. of its own speed	24.2	

Horsepower. (*Average for 3 hours.*)

Shaft horsepower of main turbines	15,876.0	15,950.0
I.H.P., main air pumps, wet, forward	12.1	9.7
after	9.8	12.4
dry	10.9	13.2
Main feed pumps, inboard, average while running	43.8	...
for three hours	14.6	66.5
outboard, average while running	...	56.1
for three hours	61.8	37.4
circulating pumps	173.0	133.0
Forced-lubrication pumps	0.45	0.5
Independent bilge pumps	2.6	1.9
I.H.P. chargeable to forced-draft blowers	363.0	
Total I.H.P. of all engineers' auxiliaries	923.0	
shaft horsepower of two main turbines	31,826.0	
horsepower of main turbines and engineers' auxiliaries	32,749.0	
Effective horsepower for bare hull	15,140.0	
Propulsive coefficient, based on bare hull	0.476	

Water Consumption.

Steam per hour, measured in tanks, pounds	224,257.0	210,172.0
chargeable to forced-draft blowers, pounds	7,125.0	
for main turbines and engineers' auxiliaries, pounds	441,554.0	
per shaft horsepower, main turbines only, pounds	210,172÷15,950= 13.18	
per horsepower of main turbines and engineers' auxiliaries, pounds	13.48	
for lighting and ventilation, pounds,	1,425.0	
for heating purposes, estimated, pounds	2,000.0	
for domestic purposes, estimated, pounds	600.0	
for steering engine, estimated, pounds	400.0	
total evaporated in boilers, pounds	445,979.0	

Coal.

Kind and quality of coal used on trial	Good quality, Pocahontas.
Heat value of coal, B.T.U. per pound	14,514.0
Total burned per hour, pounds	54,432.0
per square foot grate, pounds	37.83
per S.H.P. of main turbines, pounds	1.71
Water evaporated per pound of coal, pounds	8.20
Coal burned per twenty-four hours, tons	583.2
Knots per ton of coal	0.891

TWENTY-FOUR HOUR 12-KNOT TRIAL.

Time of beginning of trial	4:00 A. M., Nov. 6, 1909.
end of trial	4:00 A. M., Nov. 7, 1909.
Estimated average displacement during trial, tons	19,995
Corresponding mean draught, feet and inches	26-10½
Trim, inches by stern	7
Course of ship	Various.
Wind	Gentle N.W. to stiff N.N.E.
Sea	Smooth.
Average barometer reading, inches	30.62

GENERAL NOTES.

Auxiliaries in engine room were operated as shown by "Averages of Observations."

The starboard evaporators and distillers were in use for the first twenty hours, during which the required 11,000 gallons of water were evaporated. The exhaust steam from the pumps was not measured in the tanks.

The starboard feed heater was not used, owing to a leaky joint.

The auxiliary exhaust led into the port feed heater and port main condenser, and finally to the port feed tank.

The starboard measuring tanks received water from the starboard turbine only.

The port measuring tanks received water from the port turbine and from all auxiliaries whose power was required, by contract and trial-trip precept, to be used for determining water-consumption results.

Boilers E, F, G and H in fireroom No. 2 were used con-

tinuously; boiler D in fireroom No. 1 was used from 2:00 to 6:00 P. M. while cleaning fires.

Pressures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Main steam at boilers, pounds gauge	255.0	
in engine room, pounds gauge	244.0	250.7
turbine steam chest, pounds gauge	214.5	213.1
fifth-stage receiver, pounds absolute	2.99	3.03
exhaust chamber, pounds absolute	0.82	0.84
condenser, pounds absolute	0.63	0.75
Dry-air pump, steam chest, pounds gauge	43.3	36.7
Main feed line, pounds gauge	...	304.0
Auxiliary exhaust in heater, pounds gauge	...	3.9
Forced-lubrication system, pounds gauge	20.7	20.9
Engine-room bilge-pump discharge, pounds gauge	10.0	...
Dynamo-turbine steam chest, pounds gauge	208.0	...
condenser, inches mercury	28.20	...
Air pressure in fire-rooms, inches water	1.08	

Temperatures. (*Average of one-half hourly observations.*)

Main steam in engine room, degrees F	427.0	423.0
Steam at turbine chest, degrees F	406.9	416.4
superheat, degrees F	13.6	23.6
Injection, degrees F	48.1	48.5
Overboard discharge, degrees F	65.3	66.8
Heater inlet, degrees F	...	70.0
outlet, degrees F	...	153.6
Oil from main turbine bearings, mean for two bearings, degrees F	90.2	89.5
Engine-room air, degrees F	93.1	93.1
Firerooms, working level, degrees F	84.3	
Smoke stacks, by pyrometer, degrees F	549.0	

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

Main turbines	143.21	143.14
air pumps, wet, after	33.0	40.0
dry, after	255.8	277.1
feed pumps, inboard	...	19.1
outboard	...	15.1
circulating pumps	101.5	97.1
Forced-lubrication pumps	6.0	6.1
Independent bilge pumps	18.0	35.1
Ice machine, forward	...	86.0

All the auxiliaries above recorded were in continuous operation for 24 hours, except the port dry-air pump, which ran 21 hours, and the port outboard feed pump, which ran 18.5 hours.

Miscellaneous. (*Average of one half hourly observations.*)

	Starboard.		*Port.*
Number of first-stage nozzles open in main turbine..	4		4
Torsion-meter reading, average of 5-minute observations, millimeters	35.77		34.76
Torsion-meter constant	0.373		0.371
Number generators in use	1		...
Voltage at switchboard	125.0		...
Total output of generator, ampères	1,145.0		...
Power output for forced-draft blowers, ampères	484.0		...
Speed of ship, knots per hour		12.04	
Slip of propeller in per cent. of its own speed		17.50	

Horsepower. (*Average for twenty-four hours.*)

Shaft horsepower of main turbines	1,911.0		1,846.0
I.H.P. main air pumps, wet, after	2.6		3.2
dry, average while running..	...		8.9
for 24 hours.....	9.7		7.8
main feed pumps, inboard	...		31.0
outboard, average while running	...		24.7
outboard, average for 24 hours	...		19.0
main circulating pumps	19.0		17.0
forced-lubrication pumps	0.25		0.25
independent bilge pumps	1.4		3.3
ice machine, forward	...		27.6
chargeable to forced-draft blowers		85.0	
Total I.H.P. of all engineers' auxiliaries		227.0	
shaft H.P. of two main turbines		3,757.0	
H.P. of main turbines and engineers' auxiliaries		3,984.0	
Effective H.P. for bare hull		1,900.0	
Propulsive coefficient, based on bare hull		0.487	

Water Consumption.

Steam per hour measured in tanks, pounds	38,988.0		50,443.0
chargeable to forced-draft blowers, pounds		1,738.0	
for main turbines and engineers' auxiliaries, pounds		91,169.0	

	Starboard.	*Port.*
Steam per hour per shaft H.P., main turbines only, pounds.....38,988+1,911=	20.40	
per H.P. of main turbines and engineers' auxiliaries, pounds.....	22.88	
for lighting and ventilation, pounds,	2,182.0	
heating purposes, estimated, lbs.	2,000.0	
domestic purposes, estimated, pounds.....	600.0	
evaporators, estimated, pounds..	4,400.0	
steering engine, estimated, lbs...	400.0	
ice machine.....	1,132.0	
total evaporated in boilers, pounds..	101,883.0	

Coal.

Kind and quality of coal used on trial	Good quality Pocahontas.
Heat value of coal, B.T.U., per pound	14,170.0
Total burned per hour, pounds..	9,916.0
per square foot grate, pounds	23.1
S.H.P. of main turbines, pounds	2.64
Water evaporated per pound of coal, pounds	10.25
Coal burned per twenty-four hours, tons	106.2
Knots per ton of coal	2.72

TWENTY–FOUR HOUR 19–KNOT TRIAL.

Time of beginning trial	10:00 A. M., Nov. 7, 1909.
end of trial	10:00 A. M., Nov. 8, 1909.
Estimated average displacement during trial, tons	20,017
Corresponding mean draught, feet and inches	26–10⅛
Trim, inches by stern	14
Course of ship	Various.
Wind	Light N.N.E. to stiff S.W. breezes.
Sea	Smooth.
Average barometer reading, inches	30.61

GENERAL NOTES.

Auxiliaries in engine room were operated as shown by "Averages of Observations."

Four evaporators were running for the first twenty hours, during which the required 21,000 gallons of water were evaporated.

The evaporator feed pump, distiller fresh-water pump and starboard distiller circulating pump exhausted into the starboard auxiliary condenser, and the water was not measured

in tanks. The port distiller circulating pump was used on the sanitary system. It exhausted into the auxiliary-exhaust line, and the water was measured.

The starboard feed heater was not used, owing to a leaky joint.

The auxiliary exhaust was led to port feed heater and port main condenser, and finally to port feed tank.

The starboard measuring tanks received water from the starboard turbine only.

The port measuring tanks received water from the port turbine and from all auxiliaries whose power was required, by contract and trial-trip precept, to be used for determining water-consumption results. As the port distiller circulator was used on the sanitary system it was included in these auxiliaries.

Pressures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Main steam at boilers, pounds gauge	263.0	
in engine room, pounds gauge	261.9	253.6
turbine steam chest, pounds gauge	207.5	202.8
fifth-stage receiver, pounds absolute	8.9	9.2
exhaust chamber, pounds absolute	1.06	1.18
condenser, pounds absolute	0.73	0.94
Dry-air-pump steam chest, pounds gauge	51.8	43.0
Main feed line, pounds gauge	286.6	302.9
Auxiliary exhaust in heater, pounds gauge	...	4.0
Forced-lubrication system, pounds gauge	20.4	20.5
Engine-room bilge-pump discharge, pounds gauge	11.2	...
Distiller circulating-pump discharge, pounds gauge	...	3.8
Dynamo-turbine steam chest, pounds gauge	210.0	...
Dynamo condenser, inches of mercury	29.09	...
Air pressure in firerooms, inches of water	0.77	...

Temperatures. (*Average of one-half hourly observations.*)

Main steam in engine room, degrees F.	444.0	443.0
Steam at turbine chest, degrees F.	432.5	439.2
superheat, degrees F.	41.9	50.3
Injection, degrees F.	48.0	48.2
Overboard discharge, degrees F.	74.4	74.9
Heater inlet, degrees F.	...	80.3
outlet, degrees F.	...	175.6
Oil from main-turbine bearings, mean for two bearings, degrees F.	104.8	111.3

	Starboard.		*Port.*
Engine-room air, degrees F	91.8		93.0
Firerooms, working level, degrees F		85.4	
Smoke stacks, by pyrometer, degrees F		519.0	

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

Main turbines	231.94	231.94
air pumps, wet, forward	35.5	44.2
after	35.5	42.7
dry	284.5	262.7
feed pumps, inboard	30.5	32.1
outboard	30.8	32.4
circulating pumps	168.5	174.8
Forced-lubrication pumps	9.5	7.2
Independent bilge pumps	30.3	43.4
Ice machine, forward	...	80.0
Distiller circulating pump	..	30.0

All the auxiliaries above recorded were in continuous operation throughout the trial except the dry-air pumps, which ran 12 hours each, the starboard main feed pumps, which ran 16.5 hours each, the port outboard main feed pump, which ran 21 hours, and the port distiller circulating pump, which ran 20 hours.

Miscellaneous. (*Average of one-half hourly observations.*)

	Starboard.		*Port.*
Number of first-stage nozzles open in main turbine	13		13
Torsion-meter reading, average of 5-minute observations, millimeters	95.82		97.88
Torsion-meter constant	0.373		0.371
Number of generators in use	1		...
Voltage at switchboard	125.0		...
Total output of generator, ampères	1,920.0		...
Power output for forced-draft blowers, ampères	1,125.0		...
Speed of ship, knots per hour		19.21	
Slip of propeller in per cent. of its own speed		18.80	

Horsepower. (*Average for twenty-four hours.*)

Shaft horsepower of main turbines	8,290.0	8,423.0
I.H.P. main air pumps, wet, forward	5.7	7.0
after	5.7	6.8
dry, average while running	12.2	9.6
for 24 hours	5.8	4.8
feed pumps, inboard, average while running	46.7	52.5

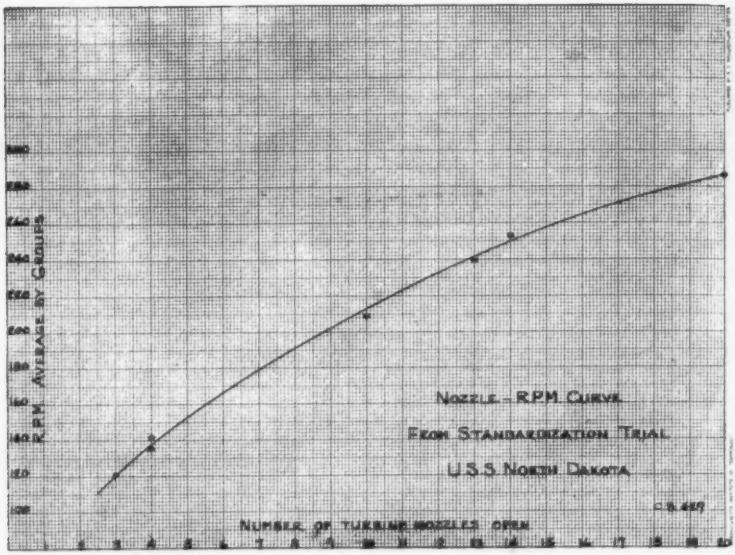
	Starboard.	*Port.*
I.H.P. main feed pumps, inboard, average for 24 hours	32.1	52.5
outboard, average while running	47.0	52.5
outboard, average for 24 hours	32.3	45.9
circulating pumps	86.0	96.0
forced-lubrication pumps	0.4	0.3
independent bilge pumps	2.8	4.4
ice machine, forward	...	26.0
distiller circulating pump, average while running	...	10.0
distiller circulating pump, average for 24 hours,	...	8.3
chargeable to forced-draft blowers	194.0	
Total I.H.P. of all engineers' auxiliaries	617.0	
shaft horsepower of two main turbines	16,713.0	
horsepower of main turbines and engineers' auxiliaries	17,330.0	
Effective horsepower for bare hull	8,300.0	
Propulsive coefficient, based on bare hull	.495	

Water Consumption.

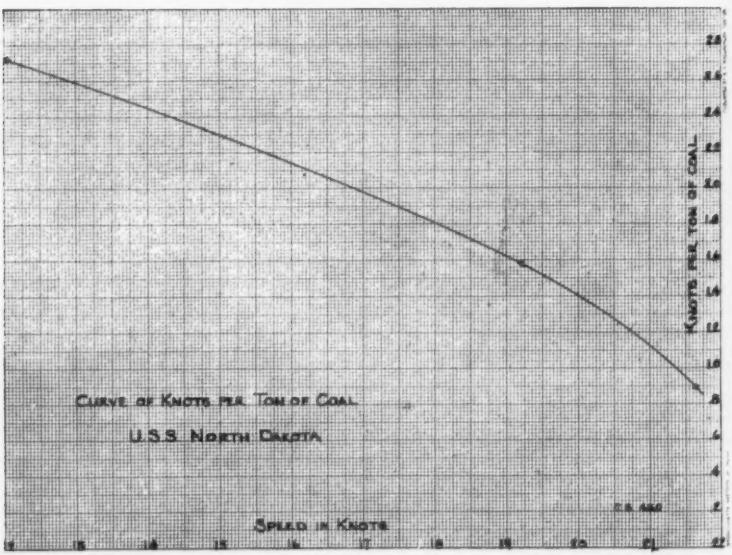
Steam per hour, measured in tanks, pounds	118,059.0	144,638.0
chargeable to forced-draft blowers, pounds	3,783.0	
for main turbines and engineers' auxiliaries, pounds	266,480.0	
per shaft horsepower, main turbines only, pounds	118,059÷8,290= 14.24	
per horsepower of main turbines and engineers' auxiliaries, pounds	15.37	
for lighting and ventilation, pounds,	2,487.0	
heating purposes, estimated, lbs.	2,000.0	
domestic purposes, estimated, lbs.	600.0	
evaporators, estimated, pounds	8,800.0	
steering engine, estimated, lbs.	400.0	
ice machine, pounds	1,017.0	
total evaporated in boilers, pounds	281,784.0	

Coal.

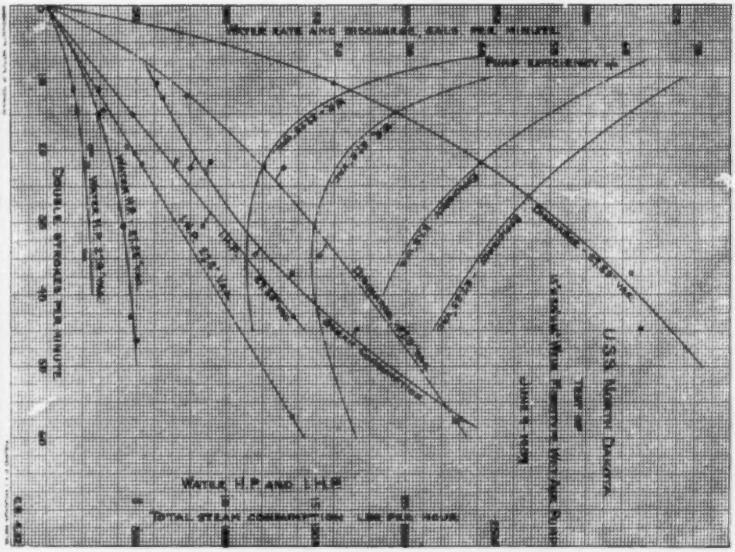
Kind and quality of coal used on trial	Good quality Pocahontas.
Heat value of coal, B.T.U., per pound	14,485.0
Total coal burned per hour, pounds	27,305.0
per square foot grate, pounds	19.0
S.H.P. of main turbines, pounds	1.63
Water evaporated per pound of coal, pounds	10.3
Coal burned per twenty-four hours, tons	292.5
Knots per ton of coal	1.58



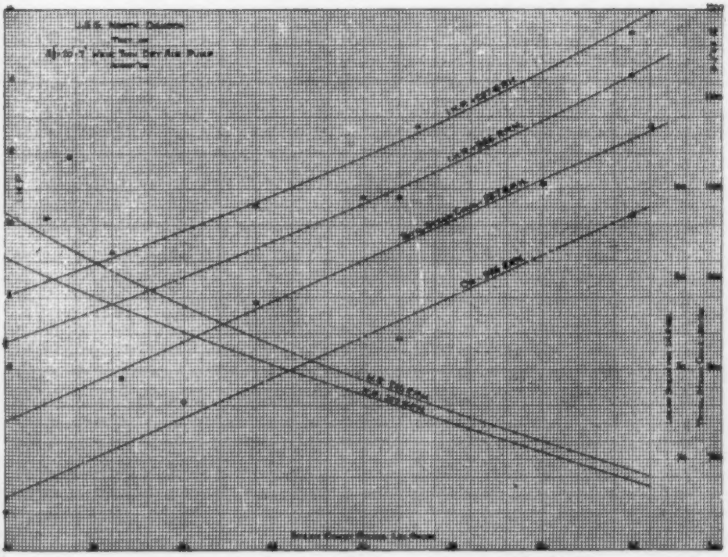
Sheet 2.—Nozzles Open and R.P.M.



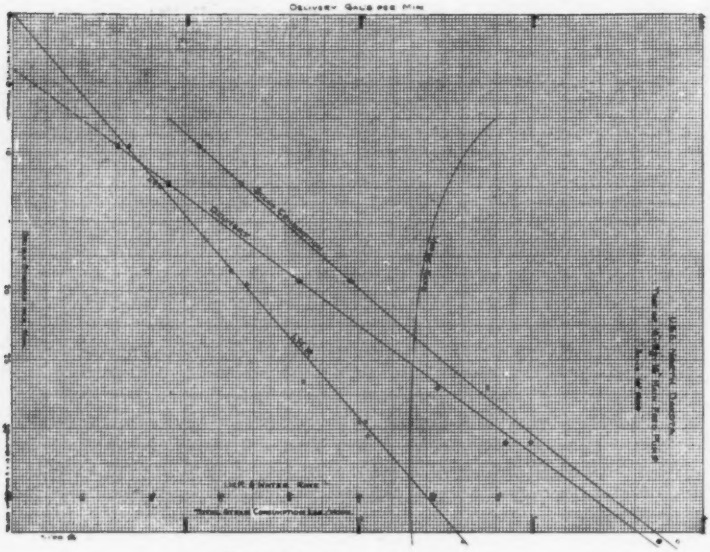
Sheet 3.—Knots per Ton of Coal.


WATER RATE AND DISCHARGE, GALS. PER MINUTE.
DOUBLE STROKES PER MINUTE
WATER H.P. AND I.H.P.
TOTAL STEAM CONSUMPTION LBS. PER HOUR
U.S.S. North Dakota
Test of
June 4, 1908

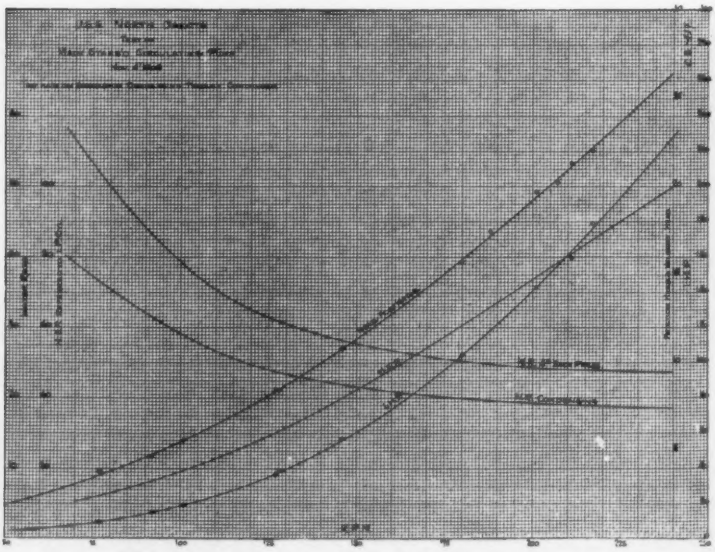
Sheet 4.—WEIR MONOTYPE WET-AIR PUMP.



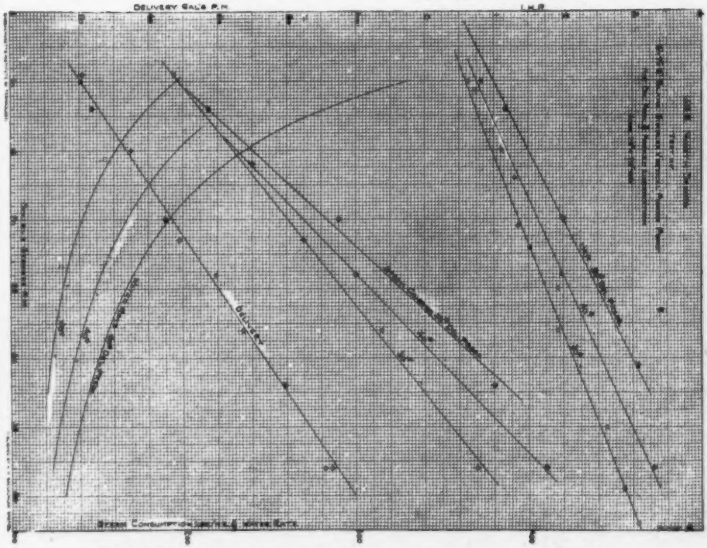
Sheet 5.—WEIR TWIN DRY-AIR PUMP.



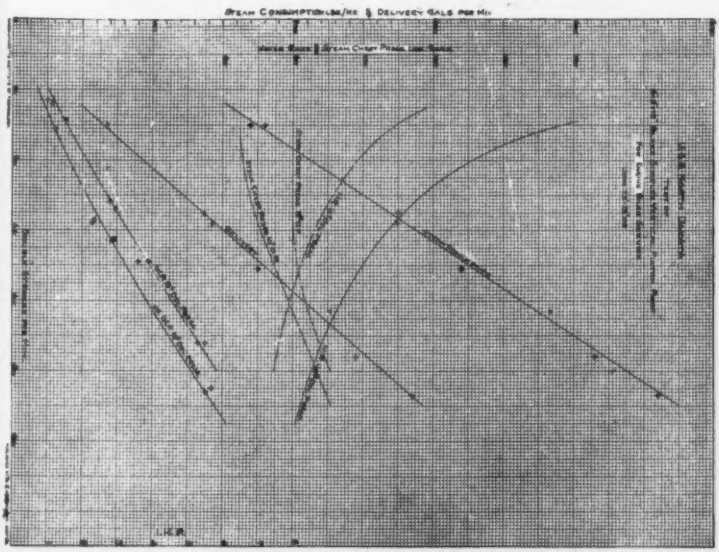
Sheet 6.—MAIN FEED PUMP.



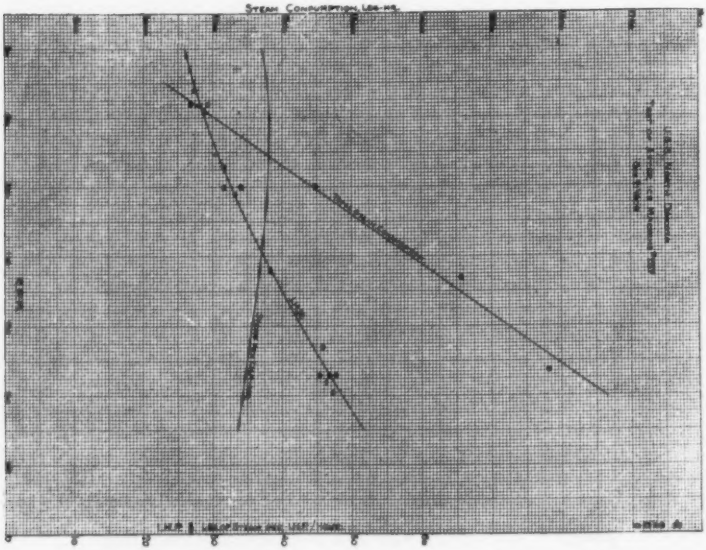
Sheet 7.—STARBOARD MAIN CIRCULATING PUMP.



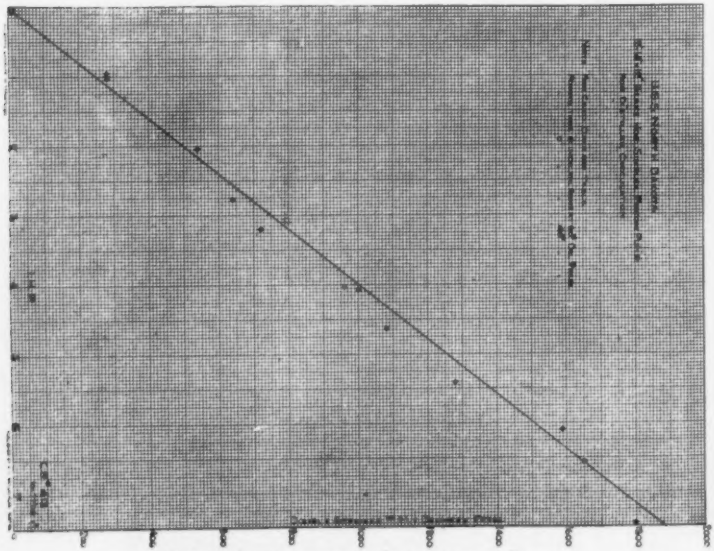
Sheet 8.—FUEL-OIL AND FORCED-LUBRICATION PUMP.



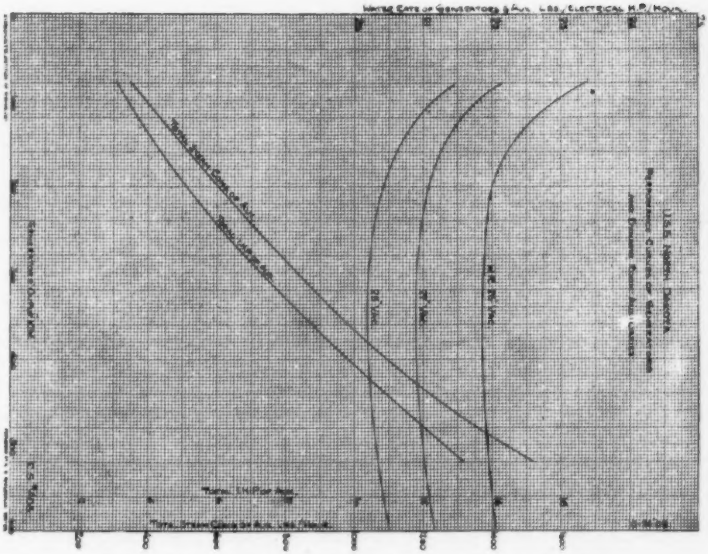
Sheet 9.—ENGINE-ROOM BILGE PUMP.



Sheet 10.—ICE MACHINE (2-TON ALLEN DENSE AIR.)



Sheet 11.—DISTILLER CIRCULATING PUMP.



Sheet 12.—GENERATORS AND DYNAMO AUXILIARIES.

U. S. S. *SOUTH CAROLINA.*

DESCRIPTION AND OFFICIAL TRIALS.

By H. C. Dinger, Lieutenant, U. S. N.

The *South Carolina* (Battleship No. 00), built by Wm. Cramp & Sons, Philadelphia, Pa., is a first-class battleship of 16,000 tons displacement, authorized by Act of Congress March 3d, 1905.

The contract was awarded July 21st, 1906; first plate laid December 18th, 1906; hull launched July 11th, 1908. The vessel was delivered to the Government November 5th, 1909, or about three years and three months from date of signing of contract.

The contract price for the construction of the hull and machinery was $3,540,000.00, of which sum $2,615,000.00 was allotted to hull and $925,000.00 for the propelling and auxiliary machinery.

PRINCIPAL DIMENSIONS OF HULL.

Length between perpendiculars, feet and inches	450–0
on load water line (24 feet 6 inches), feet and inches	450–0
over all, feet and inches	452–09
of straight keel, feet and inches	384–06
Projection forward of F.P., feet and inches	2–09
aft of A.P., feet and inches	0–0
Breadth, extreme, at L.W.L., outside of armor, feet and inches	80–02⅜
molded, feet and inches	79–10⅛
Depth, molded, main deck, at side, M.S., feet and inches	34–06⅜
upper deck, at side, M.S., feet and inches	42–10 7/32
Ratio of length to beam	5.6
Displacement per inch, at L.W.L., tons of S.W.	64.38
Area of midship section, square feet	1,890.0
L.W.L. plane, square feet	27,060.0
wetted surface, square feet	45,625.0

TANKS.

CAPACITY OF FRESH-WATER TANKS.

Reserve Feed-Water Tanks.

Compartment No.	Side.	Frames.	Cubic feet.	Gallons.	Tons F.W.
B-94	P.	56–61	814	6,066	22.6
B-95	S.	56–61	814	6,066	22.6
B-96	P.	61–66	814	6,066	22.6
B-97	S.	61–66	814	6,066	22.6
B-98	P.	66–70	650	4,858	18.1
B-99	S.	66–70	650	4,858	18.1
Totals			4,556	33,980	126.6

Trimming Tanks.

Compartment No.	Side.	Frames.	Cubic feet.	Tons S.W.	Tons F.W.
A-1	P. and S.	Stem–3	705	20.1	19.5
A-2	P. and S.	3–5	1,130	32.2	31.1
D-13	P. and S.	100–103½	1,965	56.1	54.5
D-14	P. and S.	103½-Stern post.	1,300	37.1	36.1
Totals			5,100	145.5	141.4

Fresh-Water Tanks.

Compartment No.	Tank No.	Deck.	Side.	Frame.	Gallons.	Tons.
A-55	1	Upper plat.	S.	18–22	5,060	18.8
A-55	2	Upper plat.	P.	18–22	5,060	18.8
A-55	3	Upper plat.	S.	19–22	4,215	15.7
A-55	4	Upper plat.	P.	19–22	4,215	15.7
Reservoir	1	Boat.	S.	50–52	1,004	3.7
Reservoir	2	Boat.	S.	76–77	1,004	3.7
Totals					20,558	76.4

Feed-Water and Filter Tanks.

Kind.	Location.	Side.	Gallons.
Feed.	Engine room.	P.	4,095
Filter.	Engine room.	P.	905
Feed.	Engine room.	S.	4,095
Filter.	Engine room.	S.	905
	Total		10,000

Double-Bottom Compartments.

Compartment No.	Side.	Frames.	Cubic feet.	Tons S.W.	Tons F.W.
A-94	P. and S.	11–17	2,885	82.4	80.1
A-95	P. and S.	17–22	2,256	64.4	62.6
A-96	P. and S.	22–28	2,850	81.4	79.1
A-97	P. and S.	28–32	1,951	55.7	54.1
A-98	P. and S.	32–38	2,971	84.8	82.5
A-99	P. and S.	38–43	3,818	109.0	106.0

Compartment No.	Side.	Frames.	Cubic feet.	Tons S.W.	Tons F.W.
B-85	P. and S.	43-47	3,026	86.4	84.0
B-86	P. and S.	47-52	3,765	107.5	104.5
B-87	P. and S.	52-56	2,987	85.3	82.9
B-88	P.	56-61	1,030	29.4	28.6
B-89	S.	56-61	1,030	29.4	28.6
B-90	P.	61-66	962	27.4	26.7
B-91	S.	61-66	962	27.4	26.7
B-92	P.	66-70	741	21.1	20.5
B-93	S.	66-70	741	21.1	20.5
C-97	P. and S.	70-74	2,666	76.2	74.0
C-98	P. and S.	74-78	2,544	72.6	70.6
C-99	P. and S.	78-81	1,900	54.2	52.6
D-96	P. and S.	81-84	1,880	53.7	52.2
D-97	P. and S.	84-87	1,846	52.7	51.2
D-98	P. and S.	87-93	3,720	106.2	103.3
D-99	P. and S.	93-98	2,060	58.8	57.2
Totals			48,591	1,387.1	1,348.5

MAIN ENGINES.

There are two main engines, right and left-handed, outboard turning when going ahead, each in a watertight compartment. The engines are of the vertical, direct-acting, four-cylinder, triple-expansion type, designed to develop about 16,500 I.H.P. at about 125 r.p.m., with a steam pressure of 265 pounds at the H.P. cylinder. Each engine is located in its own watertight compartment and arranged for outboard-turning shafts. The arrangement of cylinders, beginning forward, is forward L.P., H.P. and I.P., and after L.P., respectively; the F.L.P. and H.P. and A.L.P. and I.P. being bolted together, thus alowing freedom for expansion between the H.P. and I.P. cylinders in fore-and-aft and vertical planes, while a system of tie rods and braces prevent fore-and-aft and athwartship motion to the engines as a body (see Fig. 1). The material used for the cylinders is cast iron, fitted with close-grained, hard cast-iron piston and valve-chest liners, the $\frac{3}{4}$-inch space between the casings and piston liners being utilized as the steam jacket, the top and bottom heads of the I.P. and L.P. cylinders also being jacketed. Steam for the jackets is taken from the boiler side of the main-engine throttle valve, and passes successively through the cylinders, spring-reducing

F.W.
4.0
4.5
2.9
8.6
8.6
6.7
6.7
0.5
0.5
4.0
0.6
2.6
2.2
1.2
8.3
7.2
8.5

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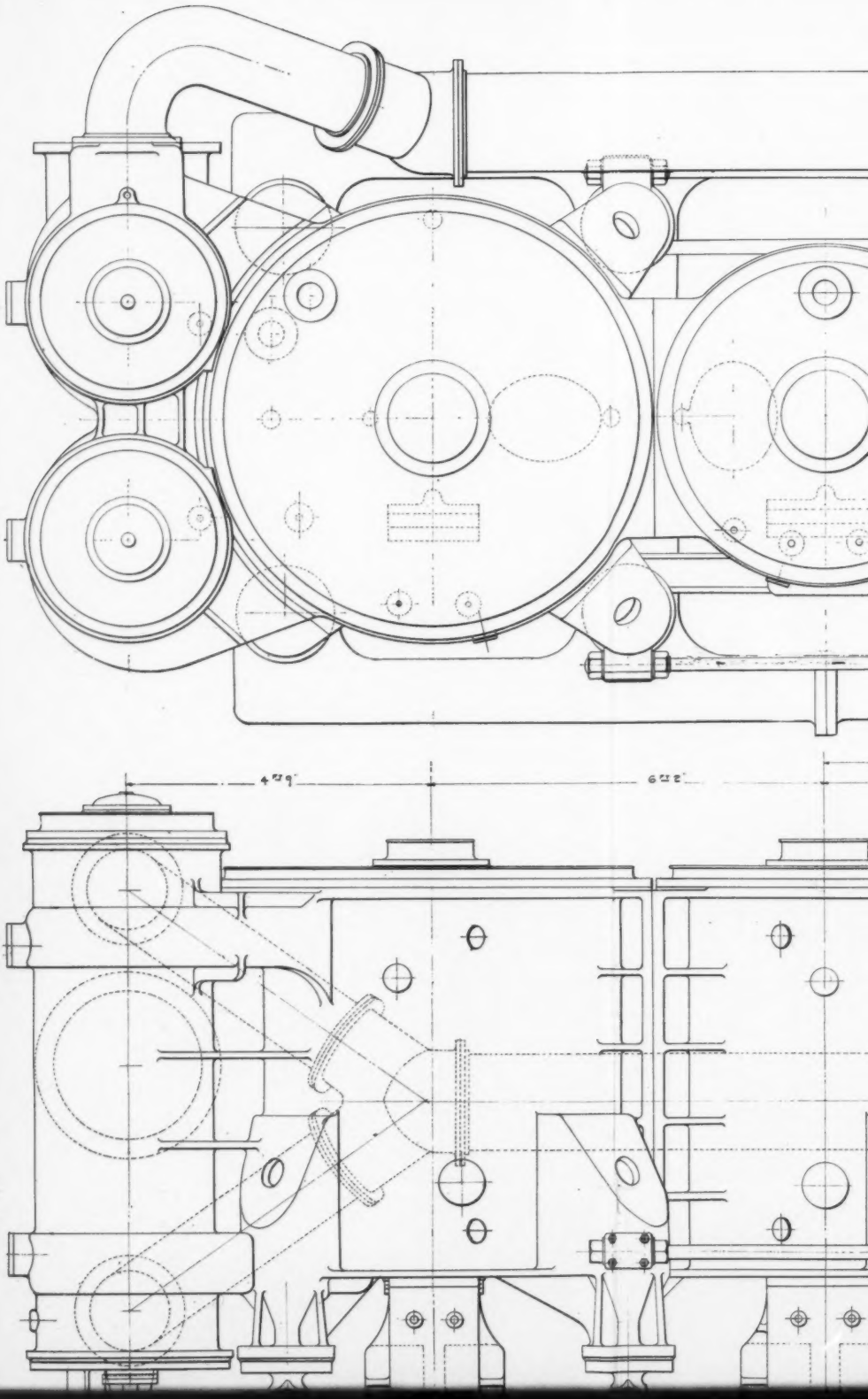
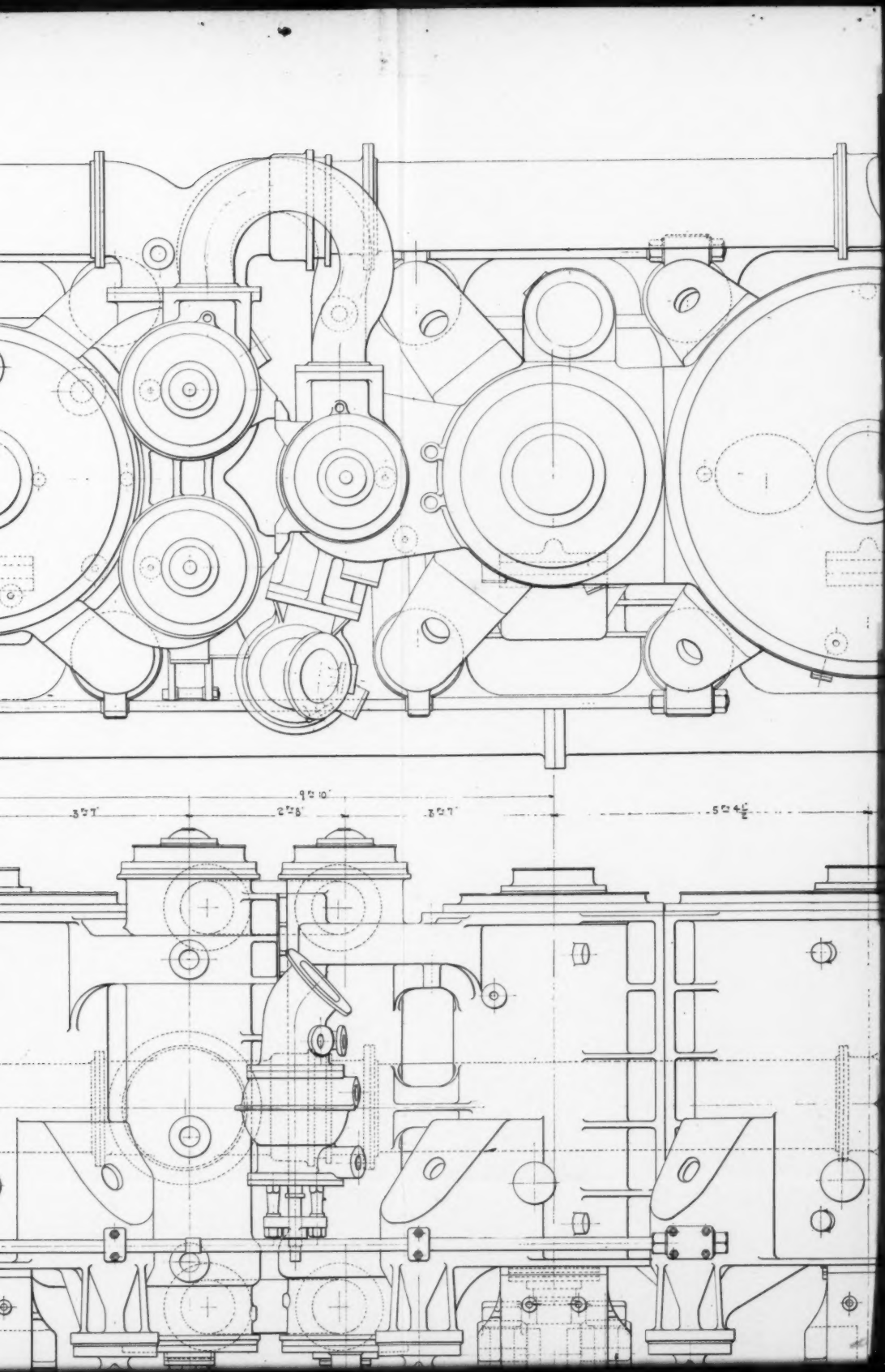
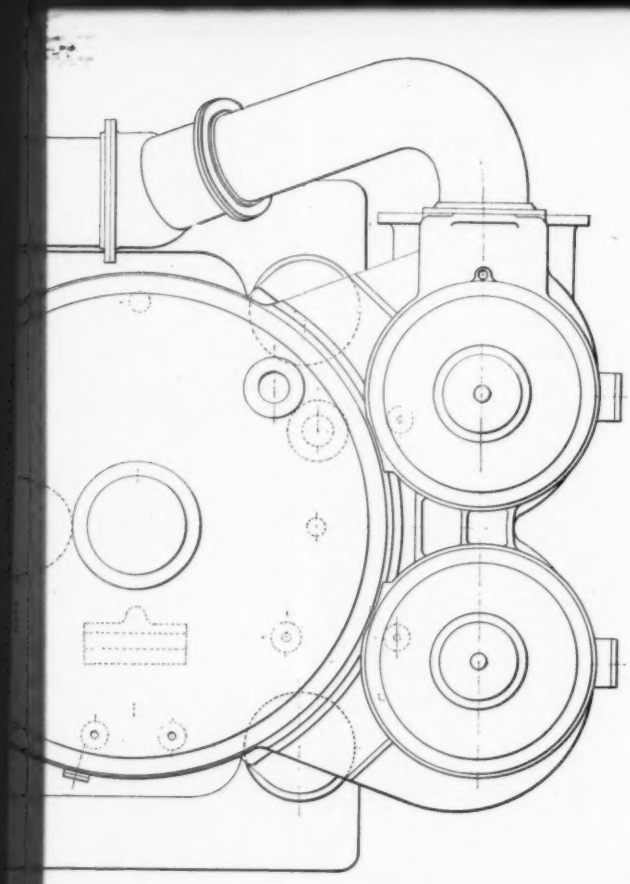
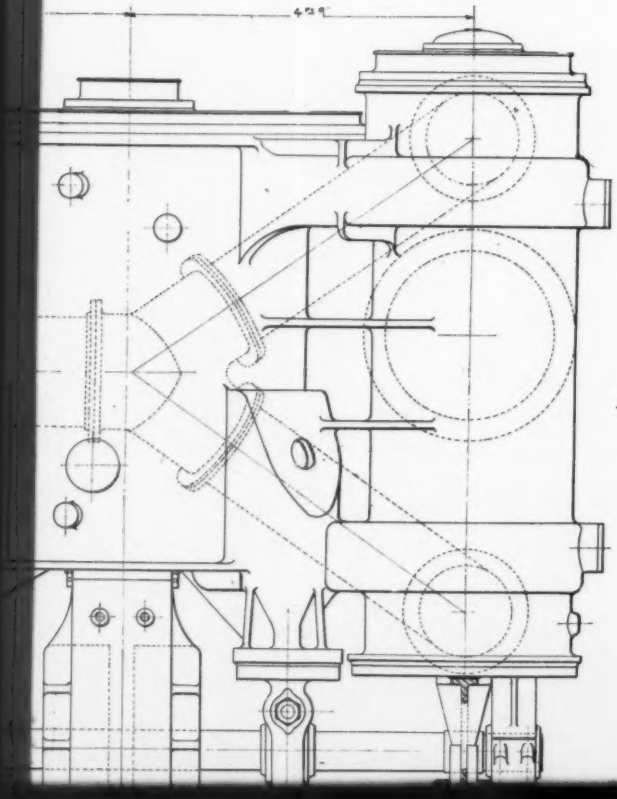
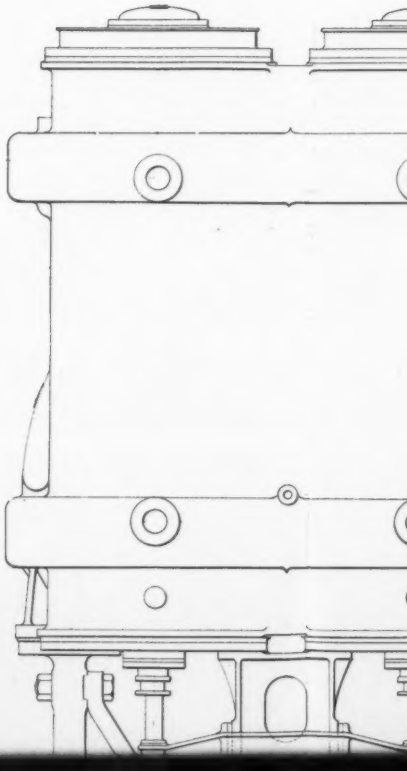
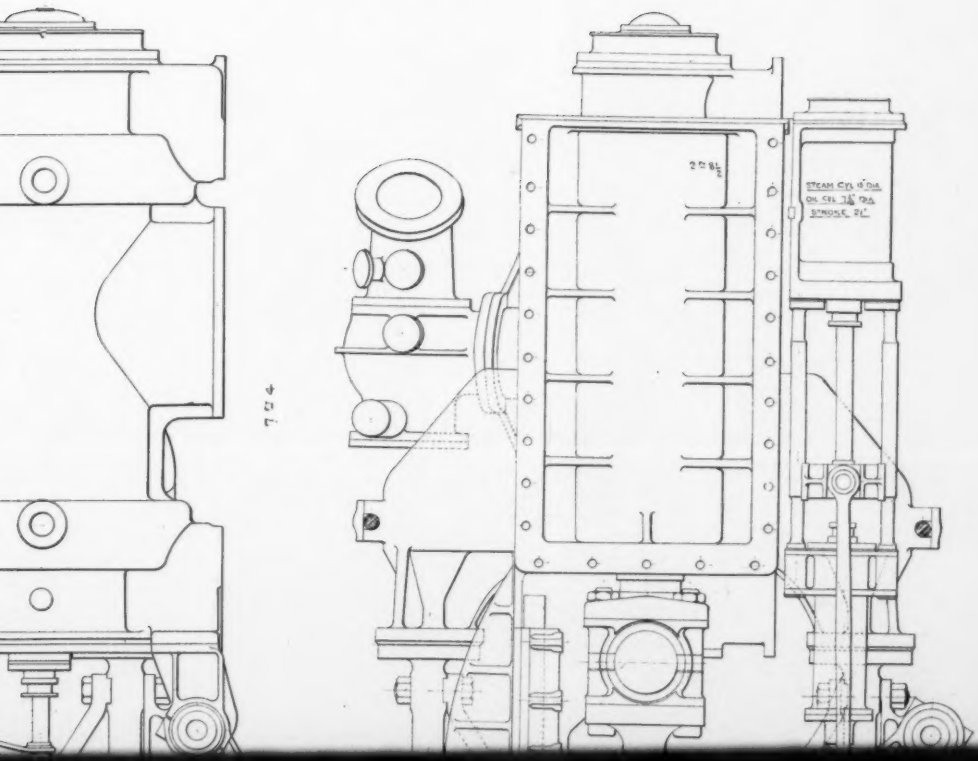


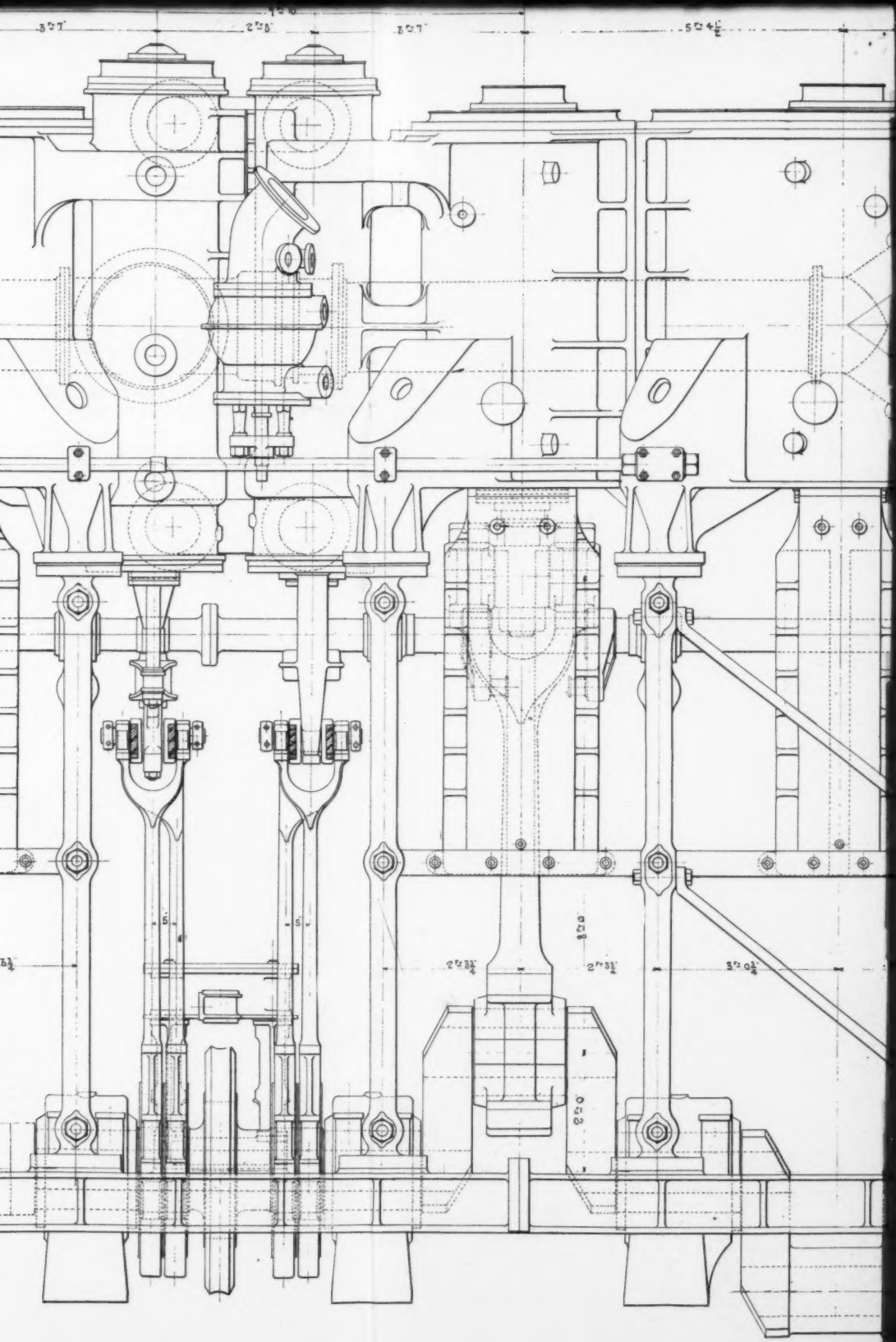




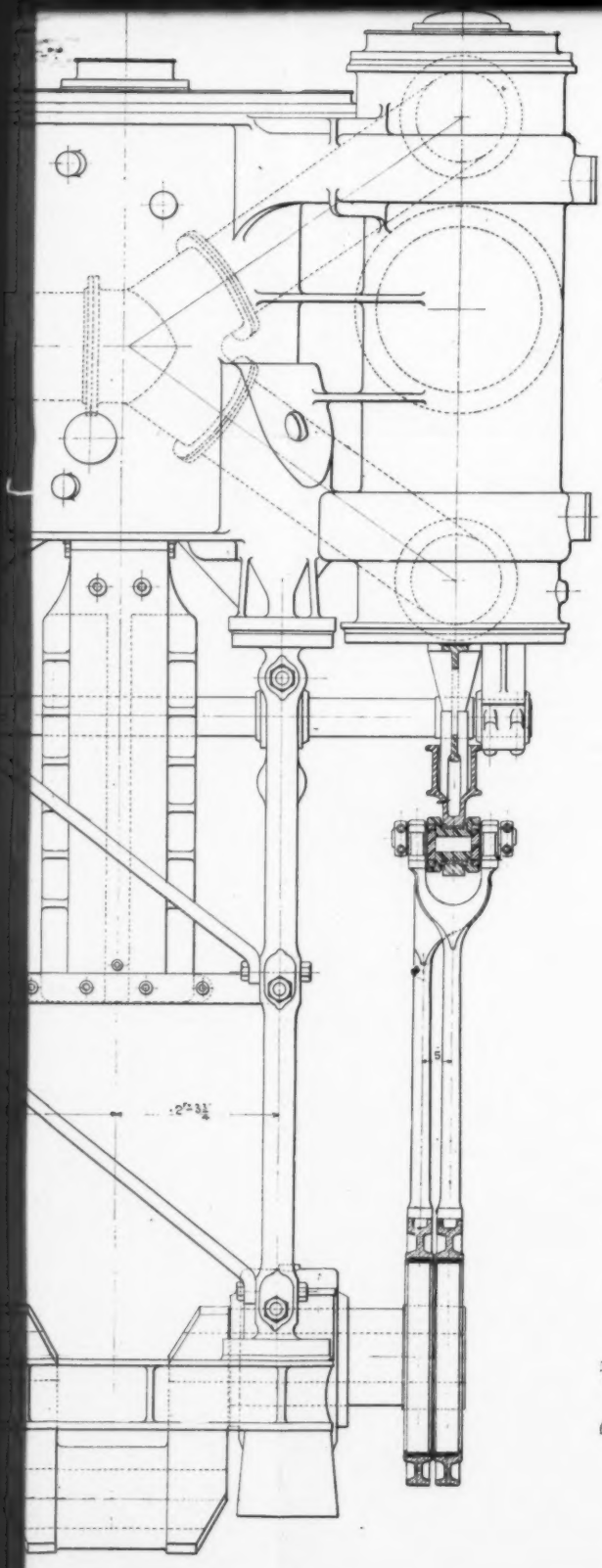


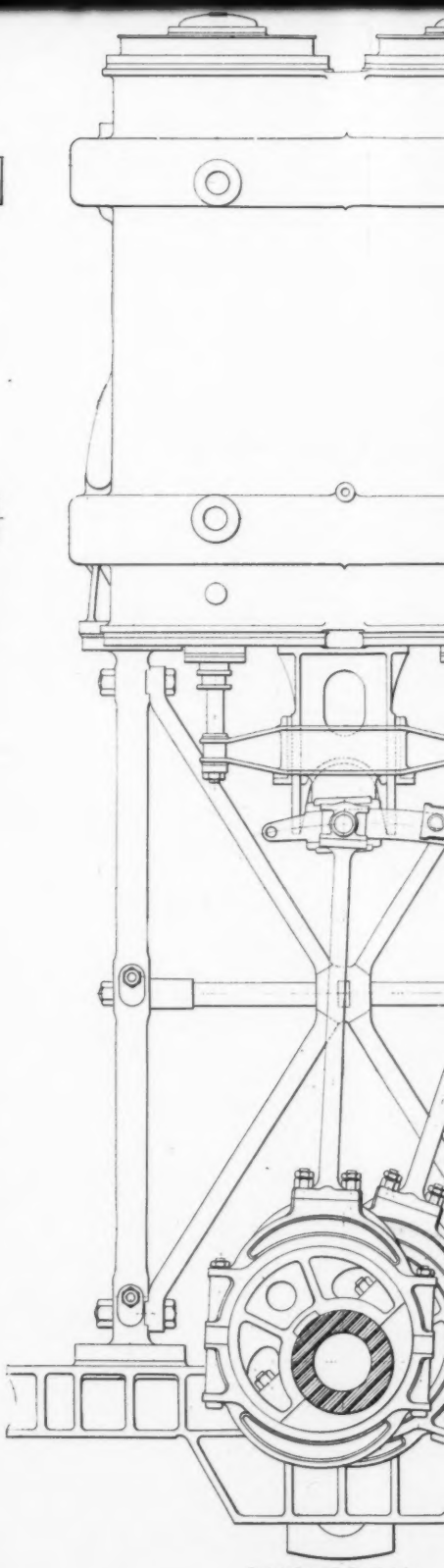



STEAM CYL Ø DIA
ON CYL 18½ DIA
STROKE 24"



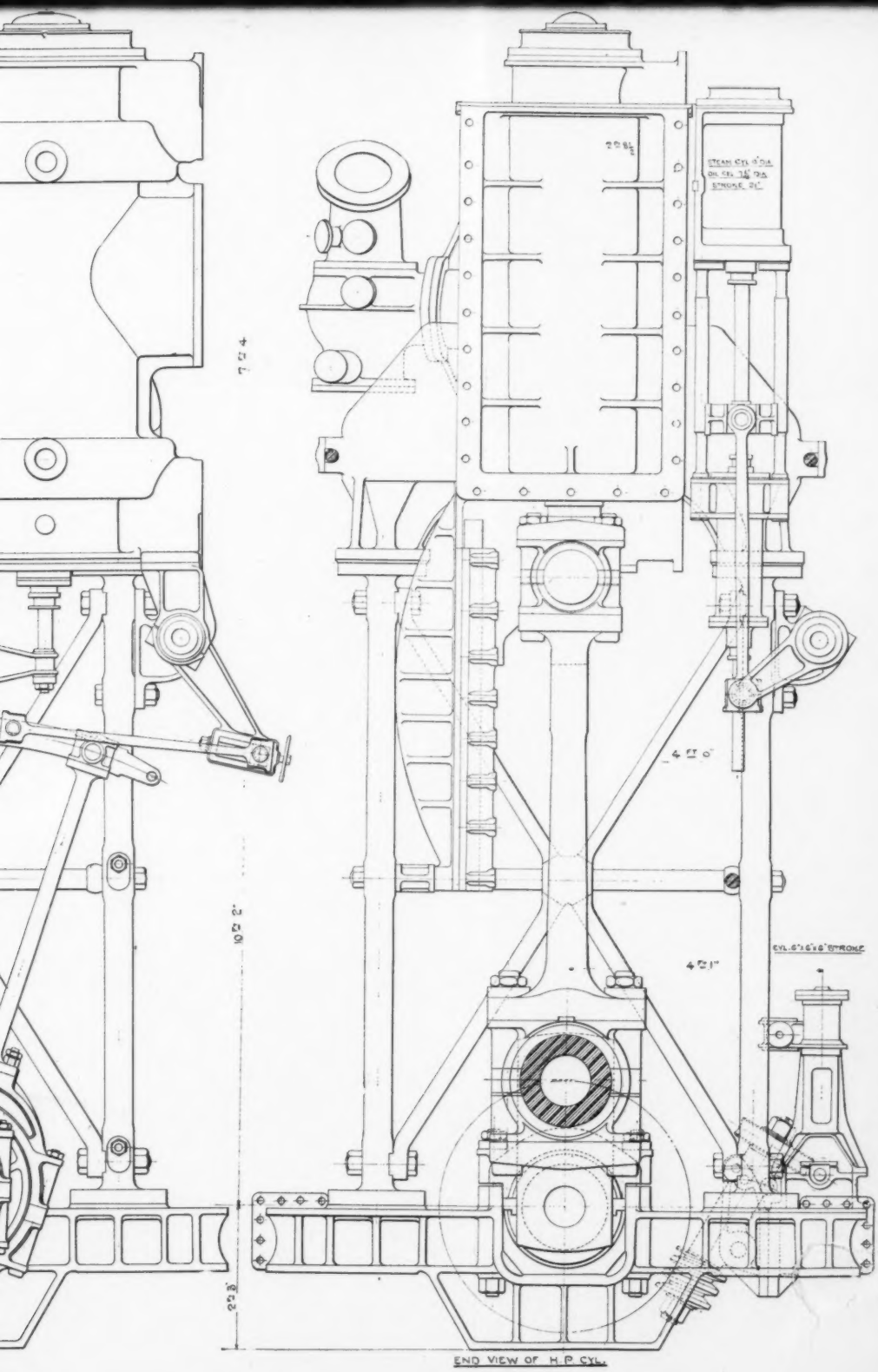
MAIN ENGINES *SOUTH*





END VIEW OF F. L. P. CYL.

SOUTH CAROLINA.



END VIEW OF H.P. CYL.

valves at the I.P. and L.P. cylinders, allowing a steam pressure of 80 to 130 and 20 to 60 pounds, respectively. Each end of the piston valves is fitted with a packing ring of hard cast iron, made practically solid, being turned larger than the bore of the valve-chest liners, then cut obliquely and the abutting ends bolted together and finished to fit the liners. The body of the valve is steel pipe with cast-steel heads. The H.P. valve stem is connected to the link block, and the I.P. and L.P. stems to a cast-steel crosshead which is connected to the link block, the Stephenson double-bar link motion with adjustable cut-off being used. The main engine valves are all fitted with balanced pistons.

The cut-offs are adjustable so as to give following:

H.P. between 0.78 and 0.44 decimal of stroke.
I.P. between 0.70 and 0.43 decimal of stroke.
L.P. between 0.60 and 0.38 decimal of stroke.

The cylinders are supported by a framing consisting of twelve forged-steel columns, $5\frac{1}{2}$ inches diameter, flanged top and bottom for bolting to the cylinders and bed plate; each athwartship pair of columns being trussed by X-braces, and stiffened fore and aft by horizontal and diagonal tie rods leading from the middle pairs of columns, thus offering no resistance to cylinder expansion.

The crosshead guides, which are of cast iron for the ahead and cast steel for the backing, are bolted to the top to facings provided on the cylinders and at the bottom to cast-steel I-section strongbacks carried by the engine framing. The guides are hollow for the circulation of water to keep them cool.

The bed plates are of cast steel, made in three sections, flanged and securely bolted together by body-bound bolts. Each bed plate when assembled consists of two longitudinal and six cross girders of I-section, well stiffened by ribs, and furnishes the seatings for the crank-shaft bearings, engine framing and turning engine. Body-bound bolts secure the bed plates to the keelsons.

The crank shaft is arranged with the cranks 90 degrees apart and is in two sections, the forward L.P. and H.P. being in one, and the I.P. and after L.P. on the other, the sequence of cranks being H.P., I.P., F.L.P. and A.L.P.

The upper ends of the connecting rods are forked for the crosshead brasses; these brasses and those of the crank pin being lined with white metal. Crank shafts and connecting rods are high-grade machinery forgings. The pistons are conical shaped, rough machined all over, the material of the H.P. being cast iron, and that of the I.P. and L.P. cast steel, the followers in each case being of the same material. The pistons are bored taper to fit the rods, and have a square counterbore on the under side to suit the shoulder on the piston rods. Pistons are secured to the rods by steel nuts, and locked in place.

The packing rings are of hard cast iron, $2\frac{1}{2}$ inches wide, being practically solid, having been cut obliquely for machining and the ends clamped solidly together; lugs cast on the back of the rings limit the play, the bore being $\frac{1}{8}$ inch greater than the corresponding bore of the piston. Springs are fitted behind L.P. rings. Adjustment for wear can be made by inserting liners between the abutting ends of rings.

The piston rods are of high-grade, and the crossheads and slippers class "A" forged steel, the slippers being faced with white metal. The ends of the rods are tapered to fit the pistons and crossheads, and the slippers are bolted to the crossheads.

The reversing engine is located outboard and is bolted to the top of the H.P. cylinder. There is an oil-control cylinder of brass connected to the steam cylinder by wrought-steel stanchions which serve as crosshead guides. The reverse shaft is also on the outboard side of the engines, near the top of the engine framing, and is connected to the crosshead of the reversing engine by two forged-steel connecting rods. Differential floating levers control the valve motion.

The turning engine is located outboard on the main-engine bed plate, between the H.P. and I.P. cylinders. The shaft carries a worm that engages a worm wheel carried by an in-

clined shaft, at the lower end of which is another worm that engages the driving worm wheel located on the crank-shaft coupling flanges. The second worm is keyed to the inclined shaft by means of a feather, but may be moved vertically to engage or disengage the worm wheel on the crank shaft.

The engines of the *Michigan* and *South Carolina* represent a decided advance in steam consumption of naval reciprocating engines. The design embodies the use of (1) superheated steam; (2) a large ratio of cylinders, namely, 1 to 2.6 to 10, which enables a good degree of expansion to be realized, and, (3) a considerable reduction in cylinder clearance. As will be seen from the engine data, the clearances are reduced to about 15 per cent. for H.P., 12.5 for I.P., and 11 for L.P. This was accomplished by taking out excess space in the steam passages and by bringing the valve ports as near as possible to end of cylinder. This later lengthens the valves considerably.

Calculations from indicator cards indicate that the steam consumption at full power was about 12.6 pounds of water per I.H.P. This result may be considered as being within a few per cent. of the actual consumption.

This shows a gain of about 16 per cent. over previous naval practice; of this gain, one-half may be assigned to the use of superheated steam and the other half due to the reduction in clearance and better cylinder proportions.

MAIN ENGINE DATA.

Cylinder Data.

	H.P.	I.P.	L.P.
Diameter, inches	32	52	72
Stroke, inches	48	48	48
Thickness of body, inches	2	$1\frac{7}{16}$	$1\frac{1}{2}$
liners, inches	$1\frac{9}{16}$	$1\frac{5}{16}$	$1\frac{9}{16}$
jacket spaces, inch	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$
valve-chest, inches	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{1}{2}$
Number of piston valves	1	2	2
Diameter of piston valves, top, inches	17	19	29
balance pistons, inches	$8\frac{1}{2}$	$7\frac{7}{8}$	$15\frac{1}{4}$
Valve travel, inches	9	10	11

Clearance:	H.P.	I.P.	L.P.
Linear, starboard, top, inch	$\frac{21}{64}$	$\frac{8}{8}$	$\frac{8}{8}-\frac{7}{16}$
bottom, inch	$\frac{37}{64}$	$\frac{9}{16}$	$\frac{23}{64}-\frac{27}{64}$
port, top, inch	$\frac{27}{64}$	$\frac{17}{64}$	$\frac{17}{64}-\frac{23}{64}$
bottom, inch	$\frac{8}{8}$	$\frac{9}{16}$	$\frac{8}{8}-\frac{37}{64}$
In per cent. of volume, starboard, top, inches	15.5	12.4	11.3
bottom, inches	16.6	13.4	12.5
port, top, inches	15.5	12.4	11.3
bottom, inches	16.6	13.4	12.5

General Engine Data.

Diameter of piston rods, inches	$7\frac{1}{4}$	$7\frac{1}{4}$	$7\frac{1}{4}$
axial hole, inches	$1\frac{1}{2}$	$1\frac{1}{2}$	$2\frac{1}{2}$
crosshead pins, inches	$10\frac{1}{2}$	$10\frac{1}{2}$	$10\frac{1}{2}$
Length of crosshead pins, each, inches	$10\frac{7}{32}$	$10\frac{7}{32}$	$10\frac{7}{32}$
slippers, inches	28	28	22
Width of crosshead slippers, inches	22	22	17
Length of backing surface, inches	28	28	22
Width of backing surface, each, inches	$8\frac{1}{4}$	$8\frac{1}{4}$	$6\frac{1}{2}$
Length of connecting rod, center to center, inches	96	96	96
Diameter of connecting rod, top, inches	$6\frac{3}{4}$	$6\frac{3}{4}$	$6\frac{3}{4}$
bottom, inches	8	8	8
hole, body, inches	$1\frac{1}{2}$	$1\frac{1}{2}$	3
crosshead end, inches	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$
pin bolts, inches	3	3	3
Number of crosshead-pin bolts	4	4	4
Diameter of crank-pin bolts, inches	4	4	4
Number of crank-pin bolts	2	2	2
Diameter of crank pins, inches	$17\frac{1}{2}$	$17\frac{1}{2}$	$17\frac{1}{2}$
Length of crank pin, inches	19	19	19
Diameter of hole in crank pins, inches	10	10 F.L.P.	10

Diameter of hole in crank pins, A.L.P., inches	6
shaft, inches	8
crank shaft, inches	$16\frac{1}{4}$
pins, inches	$17\frac{1}{2}$
shaft coupling flanges, inches	$27\frac{1}{2}$
Width of crank-shaft coupling flanges, inches	$3\frac{1}{2}$
Number of coupling bolts, each section	8
Diameter of coupling bolts, inches	$3\frac{1}{2}$
pitch circle, inches	$22\frac{1}{2}$
Width of crank webs, inches	$18\frac{1}{2}$
Thickness of crank webs, inches	10

REVERSING ENGINES.

Diameter of steam cylinder, inches	13
oil cylinder, inches	7
steam-piston rod, inches	$2\frac{7}{8}$
oil-piston rod, inches	$2\frac{7}{8}$

Stroke, inches	21
Length of connecting rods, feet and inches	4-10⅝
reverse levers, inches	18

TURNING ENGINE.

Number of cylinders	2
Diameter of cylinders, inches	6
Stroke, inches	6

SHAFTING AND THRUST BEARING.

The shafting is arranged in four sections, as follows: Thrust, line, stern-tube and propeller, all of Class A forged steel. The line and stern-tube shafts are coupled as follows: Two forged-steel collars fit on the end of the stern-tube shaft, secured to prevent turning by four keys, and into a circumferential groove between the collars near the end of the shaft is fitted a pressure ring made in four sections, the coupling bolts passing through the steel collars and the flange on the thrust shaft. The stern-tube and propeller shafts are coupled as follows: The ends of the shafts are joined by a locking ring, over which is a long clamp in halves, each half held in place on the shafts by a longitudinal key, and the two sections of the clamp coupling secured by fitted bolts.

The bodies of the stern-tube and propeller shafts are covered with composition watertight casing, and each outboard coupling is covered with a cast and wrought-steel casing filled with melted pitch.

The thrust bearing is of the horseshoe type, with a steady bearing at each end, fitted with glands to prevent the escape of oil. The body is of cast iron, so shaped as to form an oil reservoir, and the horseshoes are of cast steel, made hollow for the oil and water service. They are faced with white metal and have oil grooves on the bearing surface.

DATA FOR SHAFTING.

Diameter of thrust shaft, inches	15¼ and 15½
hole, inches	7½
Number of thrust-shaft collars	12
Outside diameter of thrust-shaft collars, inches	23½

Width of thrust-shaft collars, inches	2
space of thrust-shaft collars, inches	4
Length of thrust-shaft bearings, inches	18
Number of thrust shoes	12
Diameter of line shafting, inches	15¼
stern-tube shafting, inches	16
shaft hole, inches	8
propeller shaft, inches	16
hole, inches	8
Length of crank shaft, each, feet and inches	32–00½
thrust shaft, each, feet and inches	14–02
line shaft, each, feet and inches	25–02½
stern-tube shaft, each, feet and inches	44–06
propeller shaft, each, feet and inches	31–09½

PROPELLERS.

The hub is fitted on the tapered end of the propeller shaft and held in place by one longitudinal key and a composition nut on the end of the shaft, the nut being threaded the reverse of the direction of the propeller. The nut is locked in place and covered with a composition cap bolted watertight to the hub. The blades fit into tapered recesses in the hub, being held in place by rolled manganese tap bolts, composition chocks being fitted around the bodies of the bolts to keep the blades from shifting after being set to the desired pitch. The bolt holes in the hub are oval in shape and allow a pitch adjustment of 9 inches either way. The material for blades and hubs is manganese-bronze. The driving faces of the blades were machined to a true screw, the backing face being machined where possible and the balance of the surface ground and polished. Each propeller was accurately balanced.

DATA FOR ONE PROPELLER.

Number of blades	3
Diameter of propeller, feet and inches	16–06
Pitch for official trial, feet and inches	19–00
Pitch, adjustable, feet and inches	from 17–06 to 20–06
Ratio of diameter to pitch	1.152
Helicoidal area, square feet	84.94
Projected area, square feet	69.9
Disk area, square feet	213.8
Ratio projected to disk area	0.311

Area of immersed midship section, square feet	1,890.0
Ratio disk to immersed midship-section area	0.221
projected to immersed midship-section area	0.074
Immersion of tip of blade, inches	87.0
Tip of blade above keel, inches	9.0

DATA FROM OFFICIAL FOUR-HOUR TRIAL.

	Starboard.	*Port.*
Average revolutions per minute on course	119.43	119.49
Slip, per cent. of its own speed	11.48	11.44

BOILERS.

The boilers on this ship are of the Babcock & Wilcox type, fitted with superheaters, being identical with those of the *Michigan*. The battery consists of twelve boilers placed four each in three watertight compartments, all compartments being symmetrically arranged.

The exceptional economy of the *South Carolina* and *Michigan* is in large measure due to the use of superheaters, which operated with entire satisfaction.

The coal consumption on the full-power runs indicates an exceptionally good boiler economy. In view of the calculated water consumption of 12.6 pounds, an actual evaporation of nearly 11 pounds of water per pound of coal was obtained.

DATA FOR BOILERS.

Number of boilers	12
Length of drum, feet and inches	14–10½
Inside diameter of drum, inches	42
Thickness of drum plate, inch	1 1/16
Number of headers, each boiler	24
Heating surface per header and tubes, square feet	147.6
Total heating surface for one boiler, square feet	3,541.7
grate surface for one boiler, square feet	87.35
heating surface for all boilers, square feet	42,500.0
superheater for all boilers, square feet	4,720.0
Number of boilers per smoke pipe	6
Height of smoke pipe above grate, feet and inches	91–11½
Area through smoke pipe, square feet	77.24
Ratio grate surface to smoke-pipe area	6.786
Size of generating tubes, inches	2 and 4

Thickness of 2-inch generating tubes, B.W.G.	8.0
2-inch superheater tubes, B.W.G.	8.0
4-inch generating tubes, B.W.G.	6.0
Exposed length of generating tubes, feet and inches	9–0
Number of 2-inch generating tubes per boiler	611
2-inch superheater tubes per boiler	92
4-inch generating tubes per boiler	130
Inclination of tubes, degrees	15.0
Height of boiler, feet and inches	13–05
Length of boiler, feet and inches	13–06
Width of boiler, feet and inches	14–03½
Weight of one boiler, complete, dry, pounds	83,887
water in one boiler, at middle cock, 58° F., pounds	14,072
one boiler, complete, wet, pounds	97,909
one boiler, complete, full, wet, pounds	103,809
Total weight of boilers, dry, pounds	1,006,644
water in boilers, pounds	168,864
boilers, wet, pounds	1,175,628
Designed working pressure per square inch, pounds	295
Test pressure per square inch, pounds	450

MAIN AND AUXILIARY STEAM LINES.

The main steam pipes are arranged symmetrically in two systems, outboard, port and starboard, through the firerooms with a 7-inch cross connection at the forward end of the forward fireroom, and a 6-inch cross connection at the forward end of the engine rooms, thus completing a loop forward of the main engines. Abaft of the steam drum on the after boilers the steam lines pass over to the inboard side and through the forward athwartship bulkhead to the separators at the forward inboard end of each engine room and from these directly to the main-engine throttle valves.

The main steam pipes are supplied from each boiler drum and superheater by 5-inch pipe, the valves and connections being so arranged that the steam may be sent either through the superheater or direct, as desired. Forward of the forward boilers, in the middle boiler compartment, the size of the lines are 7-inch; for the after boilers in this compartment to the forward boilers in the after compartment the size is 10 inches, and for the after boilers to the main steam separators it is 12 inches, from which to the main engine 11-inch is used.

Steam to dynamo engines is taken, port and starboard, from

the cross connection in the forward boiler compartment, and for forward auxiliaries, heating system, galleys, whistle and siren from a casting through the necessary reducing valves on the port side. Steam for all auxiliaries in each compartment is taken from the main line; and in each engine room steam for auxiliaries (6½-inch), receivers (5-inch) and bleeder (4-inch), is taken from the head casting of the main steam separator. A 4½-inch pipe leading forward from forward cross connection supplies steam to forward auxiliaries. The main steam stop valves at each separator and the auxiliary cross connection between the engine rooms may be operated from the deck above.

Expansion is taken care of by large bends in the pipes. There are no expansion joints in main steam line in fireroom. The steam pipes are of seamless steel with wrought-steel flanges. The valves, fittings and expansion joints are of class B cast steel, with Monel-metal seats and bushings, and metallic packing for valve stems and gland stuffing boxes; all joints being scraped to a true surface, and made up metal to metal.

The auxiliary steam line in the engine rooms is taken as noted above, and forms a loop of both engine rooms through the medium of a cross-connection aft. From this line steam to the steering engine is supplied by a 4½-inch branch on the starboard side. The distributing castings in the engine rooms also supply steam to main feed pumps, main air pumps, main circulating-pump engines, engine-room fire and bilge pumps, distiller circulating pumps, auxiliary air and circulating pumps, fresh-water pumps, oil pumps, turning and reversing engines, cylinder jackets, after heating system and galley, and connections for blowing out sea chests and boiling out condensers, feed-water heaters and grease extractors.

MAIN CONDENSERS.

There is one main condenser in each engine room. They are 6 feet 9 inches in diameter by 15 feet 3 inches long by 13 feet between tube sheets. The shell plating is of ⅜-inch steel,

PUMPS AND AUXILIARY ENGINES.

Auxiliary.	Number.	Type and make.	Steam cylinders. Number.	Diameter, inches.	Diameter piston rod, inches.	Stroke, inches.	Water cylinders. Number.	Diameter, inches.	Diameter piston rod, inches.	Stroke, inches.
Main air pumps	2	Blake, twin, vertical beam	2	12	...	18	2	30	...	18
circulating pumps	2	Vertical compound engine, centrifugal pump	1 H.P. / 1 L.P.	10	...	10	1	56½ *iw*	...	...
				20	...	10	1	56½ *iw*	...	...
feed pumps	2	Blake vertical, Simplex, piston	1	18½	...	18	1	12	...	18
Auxiliary feed pumps	3	Blake vertical, Simplex, piston	1	12	...	18	1	8	...	18
Hotwell pumps, dynamo	2	Vertical, Simplex, piston	1	4½	...	6	1	5	...	6
Fire and bilge pumps, engine room	2	Blake vertical, Simplex, piston	1	12	...	18	1	10	...	18
fireroom	3	Blake vertical, Simplex, piston	1	12	...	18	1	10	...	18
Shaft and bilge pumps	2	Plunger, single-acting	...	...	...	...	1	9½	...	6
Auxiliary condenser pumps	2	Blake horizontal, Simplex, combined air and circulating	1	6	...	12	1*a* / 1*w*	10 / 10	...	12
Dynamo condenser pumps, air	2	Blake vertical, Simplex, double-acting, piston	1	9	...	12	1	18	...	12
circulating	2	Vertical, single engine, centrifugal pump	1	5	...	6	1	30*i*	...	...
Evaporating and distilling plant	1	Fresh-water, Blake vertical, Simplex, piston	1	4½	...	6	1	5	...	6
	2	Distiller, circulating, Blake horizontal, Simplex, piston	1	10	...	14	1	12	...	14
	1	Evaporator feed, Blake vertical, Simplex, piston	1	4½	...	6	1	5	...	6
Ice machine	2	2-ton Allen dense-air, with return air cooler	1	9½	...	13	1*c* / 1*e*	7 / 5½	...	13
Engine-room oil pump	2	Blake horizontal, duplex, piston	2	2	...	2½	2	1½	...	2½
Dynamo oil-cooling pump	1	Davidson horizontal, Simplex, piston	1	3½	...	6	1	3½	...	6
Air compressors	4	Westinghouse vertical	1	11	...	12	1*a*	11	...	12
Dynamo turbines	4	Horizontal Curtis 2-stage turbines, connected to double-ended generators, C. C. 4-pole, 200 kw., 1,700 r.p.m., 125 volts.								
Forced-draft blowers	12	Motor-driven Sirocco fan, air inlet (2) 23½ inches diam., fan 31 inches diam. G. E. Co. motors, 6 to 22 H.P., 650 to 1,050 r.p.m., 125 volts, capacity 17,500 cubic feet. Maximum air pressure, 2½ inches.								

a, air. *w*, water. *c*, compressor. *e*, expander. *i*, impeller.

PUMPS AND CONNECTIONS.

Pumps.	Suction pipes from—		Discharge pipes to—		Location.
	Ins.		*Ins.*		
Main air	11	Condenser.	9½	Main feed tanks.	One in each engine room.
Main circulating	18½	Sea.	18½	Condenser.	One in each engine room.
	15½	Main drain.	18½	Overboard.	
Main feed	7	Feed-tank connecting pipe.	5½	Main feed discharge main.	One in each engine room.
	5½	Channel ways.	5½	Auxiliary feed suction main.	
Auxiliary feed	3	Bottom blows.	3½	Auxiliary feed main.	One in each boiler room.
	3½	*C. & R. pipes from ship's side.	3½	*Main feed tank.	
	4	Feed suction pipe.	3½	*Reserve-feed water tanks.	
	3½	*Reserve-feed water tanks.	3	Overboard.	
Fireroom fire and bilge	5	Fireroom floors.	5	Fire main.	One in each boiler room.
	5	Sea.	5	Overboard in same compartment.	
	5	C. & R. drainage manifold.	2½	Hose connection.	
	1½	Hose connection.			
Fire and bilge	5	Sea.	5	Fire main.	One in each engine room.
	5	Engine-room bilge.	5	Overboard.	
	5	Crank pits.	2½	Hose connection.	
	4½	Shaft alley.			
	5	C. & R. drainage manifolds.			
	1½	Hose connection.			
Shaft bilge	4½	Engine-room bilges.	4½	Overboard.	One in each engine room.
	4½	Crank pits in same compartment.			
	4½	Shaft alley.			
Distiller circulating	5	Sea.	5	Distillers.	Protective deck under evaporators.
			4	Sanitary system.	
			4	Fire main.	
Auxiliary air and circulating	5½	Auxiliary condenser.	3½	Main feed tank.	One in each engine room.
	5½	Sea.	5½	Overboard.	
Dynamo air	8	Dynamo condenser.	7	Hotwell tank.	In dynamo room.
Dynamo circulating	8	Sea.	7	Overboard.	In dynamo room.
Evaporator feed	2	Sea.	2	Evaporators.	Near evaporators.
	2	Distiller circulating-water discharge			
Distiller fresh water	2	Bottom of distillers.	2	Fresh-water tanks.	In evaporator room.
			2	Main feed tanks.	
			2	Reserve-feed water tanks.	
Engine-room oil	1	Oil-storage tanks.	1	Distributing tank, or 20-gallon tanks, with overflow pipe from tank to pump suction.	One in each engine room.
Hotwell	3	Hotwell tanks.	2½	Feed suction pipe No. 1 fireroom.	One in each dynamo condenser room.
			3	By-pass to auxiliary feed pump discharge to main feed tanks.	

Connections marked * are on the after pump only.

the water chests of composition, tubes of composition—70 parts copper, 29 zinc, 1 tin; tube sheets of rolled Muntz metal, and the tube-supporting plates of sheet steel bushed with composition. The cooling surface of each condenser is 9,498 square feet, and there are 4,466 $\frac{5}{8}$-inch tubes, 16 B.W.G., each 13 feet 2 inches long.

The forward water chest, being the one for the entrance and exit of the circulating water, has a division plate on the center line of the condenser. Sizes of inlet and outlet are 18$\frac{1}{2}$ inches in diameter, the inlet nozzle being on the bottom. There are two exhaust nozzles, each 28 inches diameter.

AUXILIARY CONDENSERS.

There is one horizontal combined air and circulating pump with surface condenser in each engine room. Cooling surface 547 square feet each.

There is a condenser in each dynamo room for exclusive use of dynamo plant.

DATA FOR ONE MAIN CONDENSER.

Number of tubes	4,466
B.W.G. of tubes	16
Outside diameter of tubes, inch	$0\frac{5}{8}$
Spacing of tubes, inch	1
Length of tubes between tube sheets, feet and inches	13–0
Number of supporting plates	1
Cooling surface, measured on outside of tubes, square feet	9,498.0
Ratio of cooling surface to total heating surface	.402

DATA FOR ONE AUXILIARY CONDENSER.

Cooling surface of condenser, square feet	547.0
Outside diameter of tubes, inch	$0\frac{5}{8}$
Spacing of tubes, inch	1
Length of tubes between tube sheets, inches	92
B.W.G. of tubes	16

MAIN AND AUXILIARY FEED SYSTEMS.

Feed and filter tanks of 5,000 gallons capacity, of which 905 gallons is a filter tank, are fitted in each engine room.

Feed-Suction Pipes.—The main feed tanks in each engine room are connected by a 7-inch pipe, with a valve at the center-line bulkhead, from which on each side, with valves at the pipe, a 7-inch pipe leads to the main feed pump at the forward inboard end of each engine room, these pump-suction pipes also being cross-connected, thus forming a loop. From the suction pipe in the port engine room a 6½-inch pipe leads to the auxiliary feed pump in the after fireroom, from which to the middle fireroom it is 5½ inches, the last reduction being to 4 inches for the forward pump.

Feed-Discharge Pipes.—From the pumps in each engine room a 5½-inch pipe discharges through a grease extractor, the discharge from this connecting to and by-passing the feed-water heater, after which they unite into a common 7½-inch pipe leading to the after set of boilers, reducing to 6½ inches for the middle boilers and 5 inches for the forward set, the branches for each pair of boilers being 3½ inches, reduced to 2½ inches for each boiler; the auxiliary feed-discharge pipes being 3½ inches for each pump, reduced to 2½ inches for each boiler, both main and auxiliary feed connections at the boilers being provided with globe stop valves and screw-down check valves, the latter operated from the fireroom floor. All pipes are copper, seamless-drawn, with composition flanges, fittings and valves. Discharge pipes were all subjected to 450 pounds hydrostatic test.

The auxiliary feed pumps are also arranged so that they can each discharge into the main feed line.

EVAPORATING AND DISTILLING PLANT.

The evaporating room is located amidships at the main-deck level, between the engine-room hatches, the distillers being on the forward athwartship bulkhead of the engine hatch above the superstructure deck, and the distiller circulating pumps on the protective deck.

The plant consists of four evaporators and four distillers, the designed capacity being 16,500 gallons of potable water in twenty-four hours.

The evaporator shells and heads are of steel plate, the steam-coil heads being of composition, with the brass evaporating tubes arranged in pairs with return bends at the back end, and rolled into Muntz-metal tube sheets at the front end carried by the steam head.

Each evaporator is provided with steam gauges and relief valves for the shell and coils, also water columns, gauge cocks and blow-off valves.

The distillers are vertical and of the Bureau type. The shell is of cast iron, the covers and tube sheets of composition and tubes of brass, tinned inside and out. The tubes are expanded into tube sheets by rolling, expansion is taken care of by having the bottom tube sheet made with a flange that works in a stuffing box. The circulating water enters at the bottom, passes through the tubes and is discharged at the top; the vapor enters the shell at a point close to the top tube sheet, the water leaving at a similar point near the bottom tube sheet. Baffle plates are provided for the inlet nozzles for the circulating water and vapor.

Piping.—Steam for the evaporators and pumps is taken from a reducing valve located in the engine room from a branch on the auxiliary steam line. The vapor may be sent to the distillers or the auxiliary exhaust line, the evaporators being arranged for operation in single effect. The fresh water from the distillers passes through a small reservoir tank located in the evaporator room, from which it may drain by gravity to the main and reserve-feed tanks or may be pumped into these tanks or to the fresh-water tanks. The distiller circulating pumps have an independent connection to the sea, and, in addition to discharging to the distillers, may discharge to the sanitary system or fire main.

Oil Storage.—Lubricating-oil tanks of 2,600 gallons total capacity are fitted in engine rooms.

TURBO-GENERATING INSTALLATION.

The dynamo room of the U. S. S. *South Carolina* is located just forward of the boiler compartments, and contains four direct-current turbo-generating units. They are of the horizontal Curtis type, and were built by the General Electric Company.

Each unit consists of a 200-kw., 2-stage, 1,700-r.p.m. turbine and a direct-connected 200-kw., 125-volt, 4-pole, direct-current generator, and are of the same type and size as were described in description and trials of the *Michigan*, August, 1909, number.

The dynamo switchboard is located in a compartment on the upper platform above the dynamo room and connected electrically with the forward distribution switchboard. The dynamo switchboard controls all four turbo-generators. An additional distribution switchboard is located aft, which, together with the forward distribution switchboard, provides current to all lighting and power apparatus.

Steam is taken from the main steam line in forward boiler compartment, and there is a steam separator, port and starboard, in the dynamo room, which drains to a trap and the steam end of the air pump. The exhaust is into either the dynamo condenser, the auxiliary exhaust line or to the atmosphere. The air-pump discharge is to independent hotwell tanks, drained by an independent hotwell pump, discharging to the auxiliary feed-pump suction main.

The condensing plant consists of one independent air pump, one centrifugal pump and engine, one surface condenser, one hotwell tank and pump, in separate compartments, on port and starboard sides of the dynamo room. The sizes and pump connections are as noted in pump table. Each condenser has 1,600 square feet of cooling surface.

The lighting system comprises a total allotment of 1,500 electric fixtures, not including special outlets for signal lanterns, truck lights, Ardois signal sets and outlets for special fixtures. There are two arc lamps in each of the engine

rooms and firerooms. There are eight 36-inch searchlights, variously disposed on the superstructure.

A complete system of wireless telegraphy is installed. Besides the lighting and power system there is an interior communication system, which includes call bells, general alarms, shrill whistles, fire alarms, voice tubes, telegraphs and indicators.

VENTILATION SYSTEM.

The hull-ventilation systems have fans, delivering, combined, 111,200 cubic feet of air per minute. Artificial ventilation has been provided for quarters, living spaces, magazines, turrets, turret-trainers' station, storerooms, passages, air casings adjacent to bulkheads and decks, engine rooms, dynamo room, dynamo-condenser room, central station, waterclosets, washrooms, staterooms, around engine hatch and other enclosures where necessary to properly ventilate or insulate compartments.

The ducts have been designed to pass the number of cubic feet of air per minute through each terminal, as required, with the fan running at such speed of revolution that the actual pressure of the moving air in the duct at the fan outlet is approximately five pounds per square foot.

The air is renewed in the various spaces approximately as follows, based on the gross capacity of the compartments and on the above pressure:

Officers' quarters and crews' space, berth deck, outside of armor bulkheads, in from ten to twelve minutes.

Officers' quarters and crews' space, berth deck, inside of armored bulkheads, in from four to six minutes.

Sick-bay quarters in about eight minutes.

Watercloset spaces in about four minutes.

Storerooms in about eight to twelve minutes.

Magazines in from six to eight minutes.

Engine rooms in about two minutes.

Steering compartments in about three minutes.

General workshop in about four minutes.

Dynamo room in about three-fourths of a minute.

Dynamo-condenser room in about six minutes.

Switch and distributing-board rooms and central station in about six minutes.

The velocity of the air through terminals in quarters and living spaces is 1,000 feet per minute, sick-bay quarters 500 feet per minute, other spaces about 1,500 feet per minute, except central station, which is about 800 feet per minute.

The fans and mains, except for engine rooms, are located above the protective deck and watertight branches lead to compartments below. No ducts have been carried through the transverse slopes of the protective deck. All watertight ducts are of galvanized-steel tubing about $\frac{1}{8}$ inch thick. Natural exhaust ducts of the material mentioned above have been fitted to magazines and shell rooms. The upper ends of these ducts are carried up close to the main deck and inside of the barbettes, where practicable, and terminated with a goose neck, bell-mouthed, covered with No. 15 B.W.G. expanded metal, $1\frac{1}{2}$-inch mesh; the lower ends have bell mouths covered with the same mesh.

For ventilating the engine rooms the blowers are located below the protective deck. Branches are led from the mains to all working stations, platforms, air casing and corners of the rooms and elsewhere as required for efficient ventilation.

For ventilating the dynamo room the commutators of the dynamos are cooled by a supply of air from the ship's ventilation fans. There has been 2,000 cubic feet of air per minute delivered to each generator, and sufficient air in excess of this has been provided to change the air in the dynamo room in about two minutes instead of three-fourths of a minute as noted above.

For ventilating the turrets one fan, delivering 2,500 cubic feet of air per minute, has been provided in each turret under the upper handling room, taking its supply from the upper handling room and from the space under the upper handling room and delivering into the turret pan and trainers' station. Exhaust has been provided in the rear of the turret pan by

louvers in the mantlet plate and holes in the shelf plate. This fan places the turret pan under a low air pressure and is an auxiliary to the gas-ejecting system.

McCreery and magazine terminals, with dampers, have been fitted for all supply terminals. These terminals have been made as light as possible and are nickel-plated in quarters and galvanized elsewhere. The exhaust systems, stationary terminals have been used.

All ducts used for the ventilation of the coal bunkers are galvanized-steel tubing about $\frac{1}{8}$-inch thick, and are fitted with airtight dampers. All openings into bunkers are covered with wire mesh to keep coal out of ducts.

Especial attention has been paid to the ventilation arrangement to avoid the transmission of heat from the engine, dynamo and boiler rooms to other parts of the vessel. In wake of the boiler rooms the air casing under protective deck, and on sides of the uptake inclosures, have been connected to the ship's ventilation for supply and to the uptake enclosures for exhaust. The under side of protective deck within the engine room and the under side of the upper platform within the dynamo room have been fitted with sheathing with an air space. These air spaces are connected to the ship's ventilation for supply, and suitable exhaust provided.

DRAINAGE SYSTEM.

The drainage system consists of the following:

1. $15\frac{1}{2}$-inch main drain.
2. $5\frac{1}{2}$-inch secondary drain.
3. $4\frac{5}{8}$-inch double-bottom drain.
4. $4\frac{5}{8}$-inch forward bilge drain.
5. $4\frac{5}{8}$-inch aft bilge drain.
6. 5-inch independent drain (Bu. of S. E.).
7. 5-inch independent drain (Bu. of S. E.).

Main Drain.—It is $15\frac{1}{2}$ inches diameter throughout its entire length. It is located on the starboard side close to the coal-bunker bulkhead and extends from the after end of for-

ward fireroom to forward end of engine room, where it branches and runs athwartship, each branch connecting to the main circulating pump.

A 15½-inch stop-check valve is located in each fire and engine room, being in and forming part of main system.

All these 15½-inch stop-check valves are operated at valves and on berth deck in deck plates.

In each engine room there is a 5-inch connection between the main drain and C. and R. manifold No. 7 and No. 8, located at frame 79–80. This manifold is connected to the fire and bilge pumps in the fire and engine rooms.

Secondary Drain.—It is 5½ inches diameter throughout its entire length. It has connections to all fire and bilge pumps through the C. and R. and S. E. manifolds; also to the handy-billy pump manifold located on upper platform—forward, frames Nos. 42, 43; after, frames 80–81. The forward and after connections are made directly to the secondary drain. The forward end connects to C. and R. manifold No. 1, located in the forward fireroom at frame 44, and the after end to C. and R. manifold No. 8 in port engine room, frames 79–80.

The forward bilge drain and chain locker drain are taken by secondary drain through manifold No. 1, located in forward fireroom, frame 44.

The after bilge drain is also taken by secondary drain through manifold No. 7, located in starboard engine room, frames 79–80.

There is a 5½-inch connection from bilge well in each fireroom, and from forward and after end of each engine room.

Double-Bottom Drain.—The double-bottom drain is divided into three sections, each being 4⅛ inches diameter.

The forward section runs between 43 and 61, draining double-bottom spaces under firerooms forward of frame 56. It has a connection to C. and R. manifold No. 3, located in the middle fireroom at frame 58, also to C. and R. manifold No. 4, located in after fireroom at frame 67.

The central section runs athwartship, frames 66–67, connecting to C. and R. manifold No. 5, located in after fireroom,

starboard side, frame 67–68, and C. and R. manifold No. 6 in after fireroom, port side, frame 67–68. It drains the double-bottom compartments, outboard of the reserve-feed water tanks. This also has a connection to the forward section at frame 66–67.

The after section runs between frames 74 and 81 through the starboard engine room and drains the double-bottom compartments between frames 70 and 81. This has a connection to C. and R. manifold No. 7, located in starboard engine room, frame 79–80, which has connection to the fire and bilge pumps.

The double-bottom compartments are flooded through the S. E. manifolds connected to C. and R. manifolds by operating the stop-check lift valves in their respective manifolds.

Forward Bilge Drain.—It is $4\frac{5}{8}$ inches diameter and drains all the double-bottom ends between frames 11 and 38 (double-bottom ends at frame 11), and the hold compartments between frames 5 and 11, also forward trimming tanks forward of No. 5 bulkhead. It connects to C. and R. manifold No. 1, located in forward fireroom, frame 44, and is drained from there through secondary drain.

After Bilge Drain.—It is $4\frac{5}{8}$ inches diameter and drains all the double-bottom compartments between frames 81 and 98 (double-bottom ends at 98), and the hold compartments between frames 98–100. Also the after trimming tanks aft of bulkhead 100. It connects to C. and R. manifold No. 7, located in starboard engine room, frames 79–80, and is drained from there through the secondary drain.

Independent Drains.—These drains come under the cognizance of Bureau of Steam Engineering. In firerooms they are 5 inches in diameter and run from the bilge wells to the fire and bilge pumps. These drains are fitted in each fireroom.

The independent drains in engine rooms are 5 inches in diameter, and come under the cognizance of the Bureau of Steam Engineering. They are fitted in each engine room, and drain the bilge wells in shaft alley and engine rooms.

The drains in the shaft alley connect to C. and R. manifold

No. 7 starboard, No. 8 port, and also direct to Bureau of Steam Engineering manifold, which are connected to fire and bilge pumps in engine rooms.

C. AND R. MANIFOLDS, DRAINAGE SYSTEM.

No.	Frame.	Port or stbd.	Location.
1	44-45	P.	In forward fireroom at centerline.
2	49	P.	In forward fireroom at centerline.
3	58	P.	In middle fireroom at centerline.
4	67	...	In after fireroom at centerline.
5	67-68	S.	In after fireroom, starboard coal-bunker bulkhead.
6	67-68	P.	In after fireroom, port coal-bunker bulkhead.
7	79-80	S.	In starboard engine room at centerline.
8	79-80	P.	In port engine room at centerline.
9	42-43	S.	In passage, upper platform.
10	80-81	S.	In passage, upper platform.

TRIMMING TANKS—PUMPING AND DRAINAGE.

There are four trimming tanks, two forward and two aft. Those forward are flooded by a 5-inch sea connection, frame 5–6, starboard side.

A 3-inch pipe is led to each trimming tank, which is controlled by a 3-inch globe valve, operated at valve and on berth deck in deck plates, frame 5–6.

Those aft are flooded by a 5-inch sea connection, frame 100–101, starboard side. A 3-inch pipe is led to each trimming tank, which is controlled by a 3-inch valve, operated at valve and on berth deck in deck plates.

Draining.—The forward tanks are drained through the forward bilge line to manifold No. 1, frame 44, and the after tanks by the after bilge to manifold No. 7, frame 78–79, starboard engine room.

The extreme forward and after tanks are sluiced into adjacent compartments.

From these manifolds Nos. 1 and 7 water is handled by the secondary drain.

DESCRIPTION OF FIRE MAIN.

The fire main is composed of two systems, 6 inches diameter, port and starboard, located near center line of ship, between frames 43–81, at the forward and aft end; between

frames 43–44 and 80–81, the main is run athwartship, thus forming a complete circuit between frames 43 and 81 ; extensions from the circuit are made forward and aft.

FRESH–WATER SYSTEM.

From the distiller pump the distilled water is discharged to the fresh-water tanks on the upper platform forward. From these tanks the water is pumped through the distributing main into the reservoir tanks on the boat deck by use of the electric pumps.

All spaces are supplied with fresh water from the reservoir tanks by gravity, and, in order that all spaces may be supplied in case the gravity tanks are out of commission, the electric pumps are so arranged that they can be run continuously, the surplus water being returned to the suction through relief valves fitted in the discharge pipes.

The washrooms for chief petty officers, firemen, servants and the laundry are fitted with storage tanks, from which the fixtures are supplied, while all other spaces are supplied directly from the distributing main or its branches.

For filling the ship's tanks from the deck or water boat, connections are provided near frame No. 30 with a by-pass to the fresh-water main. The filling pipes for the reserve-feed water tanks near frame No. 71 is also by-passed to the fresh-water main.

On the upper deck, between frames No. 43 and No. 44, port side, a 1½-inch hose connection is fitted for supplying fresh water to the steam cutters.

SALT–WATER SYSTEM.

The sanitary or flushing system is supplied by a direct connection from the distiller circulating pump and a by-pass from the fire main.

All the plumbing spaces except the crew's washroom and the crew's head are supplied by this main, and a branch is run to the circulating pump in the ice-machine room.

The crew's washroom and the crew's head are supplied by an independent system, through two electrically-driven centrifugal pumps, drawing from an independent sea valve. This system, for crew's spaces, is also fitted with a by-pass from the fire main.

HEATING SYSTEM.

All pipes for the heating system consist of seamless-drawn brass pipe, iron-pipe size; fittings, flanges and valves of composition.

Radiators, consisting of pipes led along deck, are made up of 2-inch pipe; those installed on bulkheads consist of coils, made up of 1-inch pipe. Radiators in crew's spaces are fitted with key valves, those in officers' quarters with valves having fixed hand wheels; valves being provided with union on outlet end so that radiator can be taken down without disturbing the lines. Radiators are divided into two or more sections where the area exceeds eleven or more square feet, each section being provided with independent steam, drain and air valves to avoid the necessity of having steam on the entire heating surface during mild weather.

All radiator pipes, fittings and valves are of polished brass; where the size does not exceed 1½ inches the connections for circuit steam and drain pipes are made with screwed fittings, but above that size flanges are used.

Watertight connections at decks and bulkheads are made with composition stuffing boxes and copper U-bends installed throughout the lines to provide for expansion.

The steam supply to radiators is divided into two systems, one forward and one aft, and is taken from the auxiliary steam lines in the fire and engine rooms respectively, the connections at the auxiliary lines being fitted with pressure gauges, stop, relief and reducing valves. The steam is then carried to distributing manifolds on the berth deck and divided into circuits, each circuit being fitted with a stop valve so that it can be operated independently.

The steam supply to galleys, pantries, bakery and plumb-

ing heaters is likewise taken from the auxiliary lines, fitted with gauges and valves and divided into two systems, one forward and one aft. The exhaust from radiators is also divided into circuits and led to receiving manifolds, each circuit having a stop and check valve to prevent water from one circuit backing up into another. The discharges from manifolds are led to traps in engine rooms, and the drains from these traps are connected together and lead to drain manifolds, and discharge directly to port or starboard filter tank or to main or auxiliary condensers. The drains from the plumbing heaters have no connection whatever with other drains, and are carried directly to the auxiliary exhaust lines.

The steam to radiators has been arranged to work at 50 pounds, and the steam to galleys, pantries and water heaters at 30 pounds per square inch.

All systems are tested at 150 pounds per square inch.

ASH HOISTS.

The ash-hoist engines, six in number, were designed and built by builders. They are located in the upper hatches, at gun-deck level, the ventilator trunks containing the bucket guides, etc.

The hoisting engines are of the reciprocating type, with a rope drum, fitted with a follow-up and reversing gear, and an adjustable safety gear to prevent overwinding and stop the engine when the ash bucket reaches the fireroom floor.

FORCED–DRAFT BLOWER.

The forced-draft blowers are located over the firerooms, outside the hatch enclosure. The motors were designed and built by the General Electric Company, and are of a modified Sirocco type.

The arrangement is the same for all of the firerooms, there being four motors, each driving one fan, located outboard, and so that the air is discharged directly to the fireroom about over each boiler. The supply is taken from the fireroom-

ventilator ducts, which are closed at the bottom when under forced draft.

The motors are designed for speeds varying from 650 to 1,050 r.p.m., developing from 6 to 22 H.P., and are wound for 125 volts.

The fans are of the double-inlet Sirocco type, 31-inch diameter, each with a capacity of 17,500 cubic feet of air per minute at 2½-inch air pressure.

MACHINE SHOP.

The machine shop is located amidships on the berth deck forward of the after 12-inch barbette and between the machinery hatches, access to it being from the athwartship passage and from the engine rooms through their respective entrance hatches. The machines are arranged with independent enclosed motors.

The following machines are installed:

1 25-inch extension gap lathe;
1 12-inch tool-room lathe on bed of gap lathe;
1 14-inch engine lathe;
1 16-inch crank shaper;
1 31-inch radial-drill press;
1 16-inch sensitive-drill press;
1 30-inch grindstone;
1 Universal milling machine;
1 12-inch emery grinder;
1 Combined hand punch and shears;
6 Bench vises;
1 Portable boring machine.

The tools are provided with the most modern attachments, including scroll and drill chucks, index head, automatic cross feed, swivel table, pipe vises, etc., and all necessary tools, drills and cutters.

AIR-COMPRESSOR PLANT.

There are four 11-inch Westinghouse steam-driven air compressors, two in each engine room. The plant is designed to have a capacity of 188 cubic feet of free air per minute at 125 pounds air pressure. There is an air tank of 61.05 cubic feet capacity.

This plant is designed to furnish air for operating pneumatic tools, for blowing soot off boiler tubes and for use of smoke-ejecting system of battery.

A copper main is run through machinery space, with outlets convenient to the boilers.

OFFICIAL TRIALS.

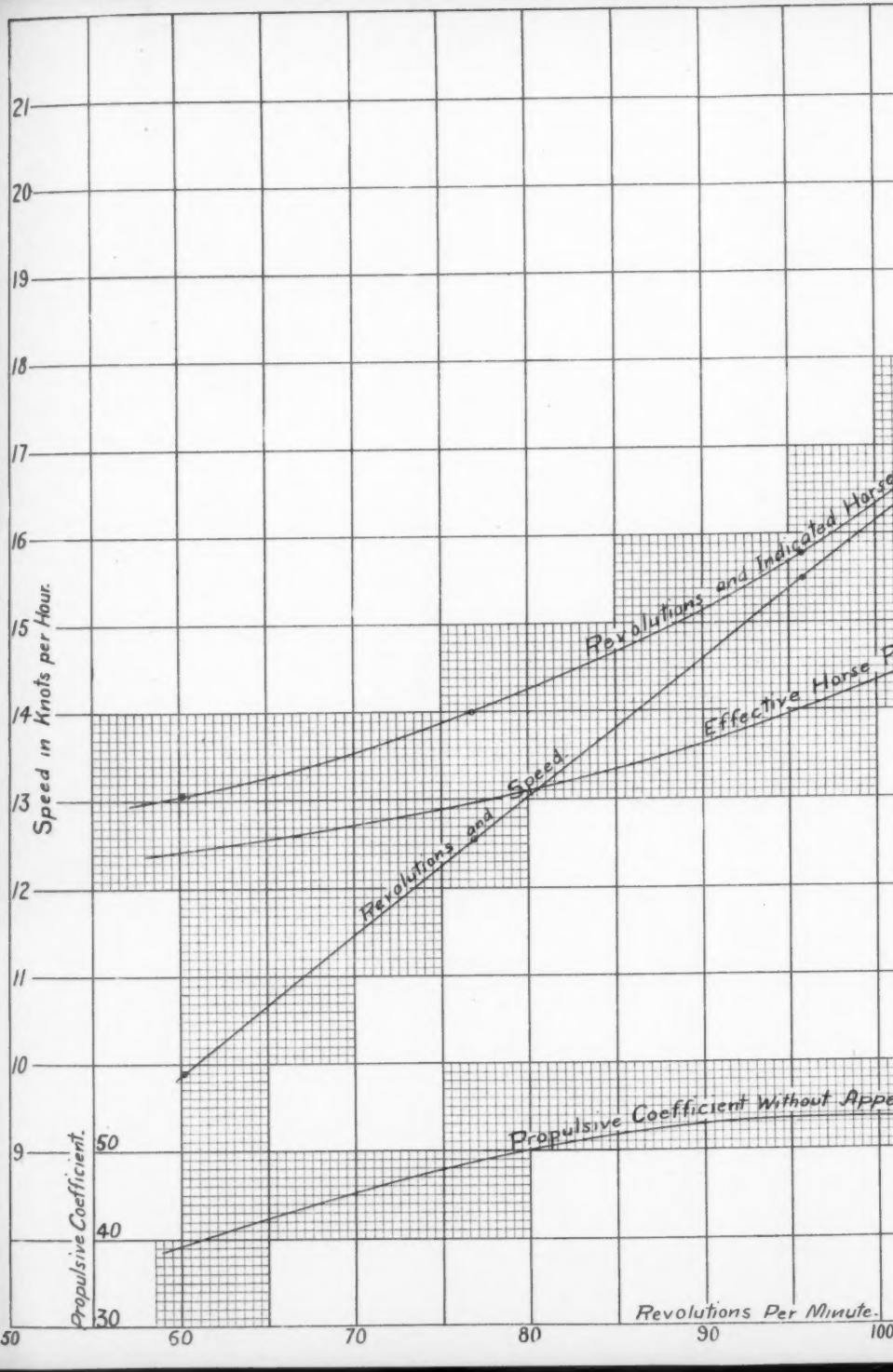
The contract required an average speed of not less than 18½ knots in the open sea for four consecutive hours, at a mean draught of 24 feet 6 inches, corresponding to a displacement of 16,000 tons; and twenty-four-hour endurance and coal-consumption trials under the same conditions at 17½ and 12 knots speed.

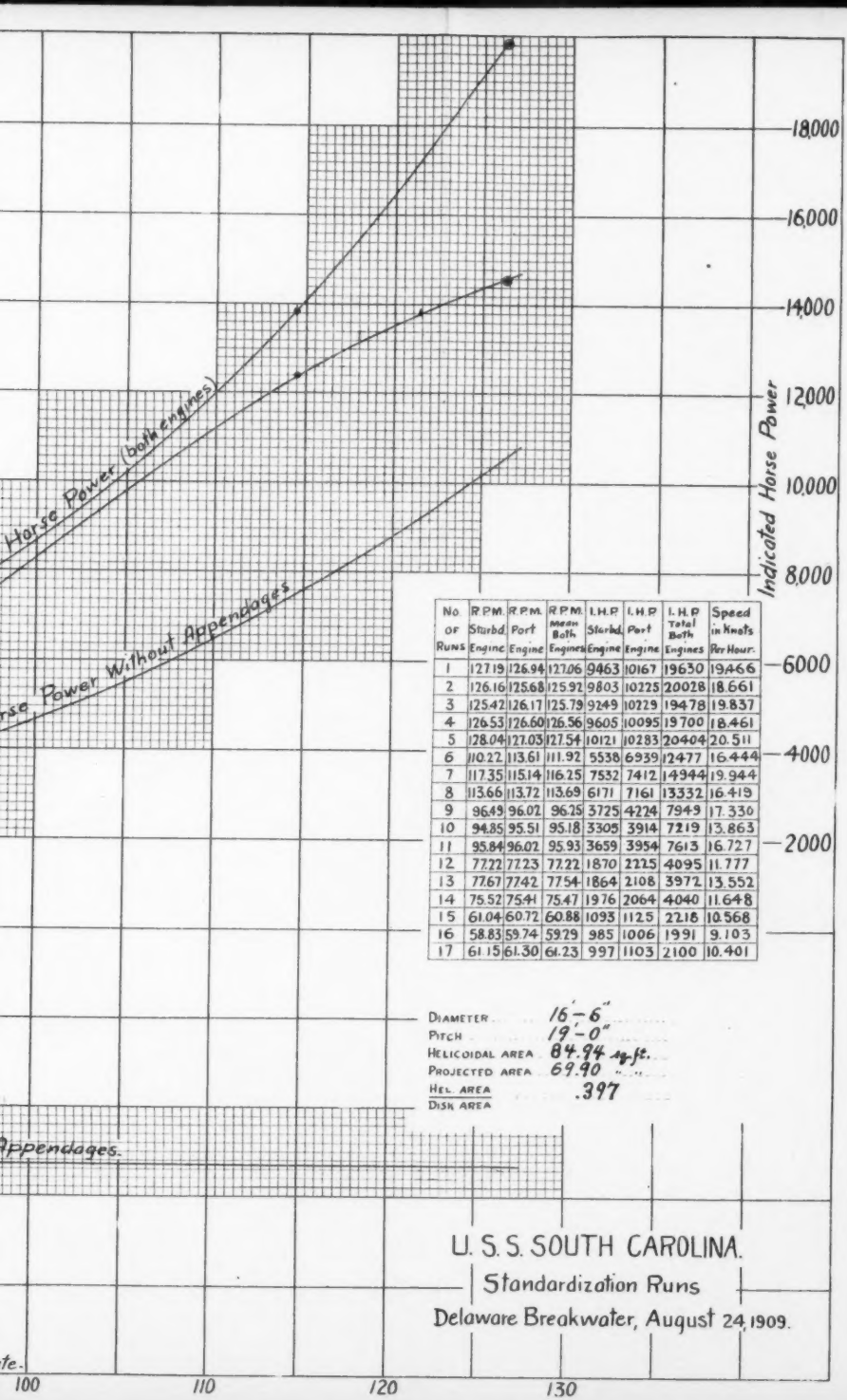
The standardization trial took place off Delaware Breakwater on August 24, 1909. The revolution-horsepower curve accompanying was constructed from the data obtained. Some of the principal engineering data recorded on standardization runs are shown in accompanying tables.

Draught at standardization: Forward, 23 feet 11⅛ inches; aft, 25 feet 3⅛ inches; mean, 24 feet 7⅛ inches. Corresponding displacement, 16,064 tons.

FOUR-HOUR FULL-SPEED TRIAL.

This trial began at 8:30 A. M., August 25, 1909, off mouth of Delaware Bay. The weather was fair, with light to gentle breezes from S.W. and moderate sea from S.W. The courses steered were S. by W. ⅛ W. and N. by E. ⅛ E., each for two hours. The following are the data for this trial:


Speed in Knots per Hour.
21
20
19
18
17
16
15
14
13
12
11
10
9
Revolutions and Indicated Horse
Effective Horse P
Revolutions and Speed
Propulsive Coefficient Without Appe
Propulsive Coefficient.
50
40
30
Revolutions Per Minute.
50
60
70
80
90
100



No. of Runs	R.P.M. Sturbd. Engine	R.P.M. Port Engine	R.P.M. mean Both Engines	I.H.P. Sturbd. Engine	I.H.P. Port Engine	I.H.P. Total Both Engines	Speed in Knots Per Hour.
1	127.19	126.94	127.06	9463	10167	19630	19.466
2	126.16	125.68	125.92	9803	10225	20028	18.661
3	125.42	126.17	125.79	9249	10229	19478	19.837
4	126.53	126.60	126.56	9605	10095	19700	18.461
5	128.04	127.03	127.54	10121	10283	20404	20.511
6	110.22	113.61	111.92	5538	6939	12477	16.444
7	117.35	115.14	116.25	7532	7412	14944	19.944
8	113.66	113.72	113.69	6171	7161	13332	16.419
9	96.49	96.02	96.25	3725	4224	7949	17.330
10	94.85	95.51	95.18	3305	3914	7219	13.863
11	95.84	96.02	95.93	3659	3954	7613	16.727
12	77.22	77.23	77.22	1870	2225	4095	11.777
13	77.67	77.42	77.54	1864	2108	3972	13.552
14	75.52	75.41	75.47	1976	2064	4040	11.648
15	61.04	60.72	60.88	1093	1125	2218	10.568
16	58.83	59.74	59.29	985	1006	1991	9.103
17	61.15	61.30	61.23	997	1103	2100	10.401

Diameter ..... 16'-6"
Pitch ..... 19'-0"
Helicoidal Area ..... 84.94 sq. ft.
Projected Area ..... 69.90 " "
Hel. Area / Disk Area ..... .397

Appendages.

U. S. S. SOUTH CAROLINA.
Standardization Runs
Delaware Breakwater, August 24, 1909.

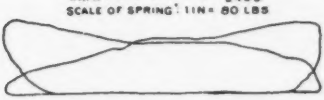
STARBOARD ENGINE.

H. P. CYLINDER.

M.E.P. 100
I.H.P. . . . . 2330.
SCALE OF SPRING : 1IN. = 150 LBS.

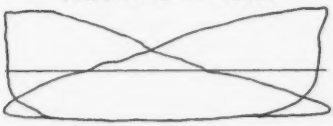
I. P. CYLINDER.

M.E.P. 39
I.H.P. 2439
SCALE OF SPRING : 1IN = 80 LBS.



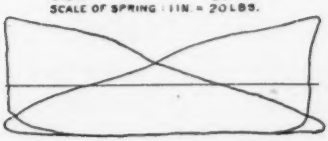
F. L. P. CYLINDER.

M.E.P. . . . . 18.5.
I.H.P. . . . . 1867.
SCALE OF SPRING : 1IN. = 20 LBS.



A. L. P. CYLINDER.

M.E.P. . . . . 17.
I.H.P. . . . . 2048.
SCALE OF SPRING : 1IN. = 20 LBS.



INDICATOR CARDS MAIN ENGINE "SOUTH CAROLINA." ONE HALF ACTUAL SIZE.

STARBOARD ENGINE DATA (ONE SHEET FOR EACH ENGINE), STANDARDIZATION TRIALS.

(Fill in with india ink for blueprinting.)

U. S. S. SOUTH CAROLINA, COURSE OFF DELAWARE BREAKWATER, ON AUGUST 24, 1909.

Number of run or observation.	Time. H.	M.	S.	Reading of engine counter.	Revolutions per min.	Steam pressure. At engine, per gage.	H. P. steam chest per gage.	I. P. receiver abs.	L. P. receiver abs.	Vacuum in inches, mercury.	Barometer.	Cut-off in cyl. in decimal of stroke. H.P.	I.P.	F. L. P.	A. L. P.	Temperature. Outside air.	Engine room working level.	Injection.	Discharge.	Hot well.	Feed (entering boilers).	Mean effective pressure. H. P. cyl.	I. P. cyl.	F. L. P. cyl.	A. L. P. cyl.	Referred pressure.	Indicated horsepower. H. P. cyl.	I. P. cyl.	F. L. P. cyl.	A. L. P. cyl.	Total.
1					127.19	295	250	97	37	26.5		.79	.72	.45	.44		102	72		108	180	110.5	41	15.9	17.2	37.8	2670	2659	1986	2148	9463
2					126.16	275	245	100	37.5	26.5		"	"	"	"		102	72		108	172	108.5	46.5	16.5	17.5	39.6	2600	2991	2044	2168	9803
3					125.42	300	245	95	36.5	26.5		"	"	"	"		101	72		106	180	112.75	38.75	16	17.75	37.55	2675	2438	1971	2125	9249
4					126.53	293	252	100	37.5	26.5		"	"	"	"		102	72		106	175	103	45.5	16	17.75	38.65	2476	2935	1988	2206	9605
5					128.04	290	257	102	39.5	26.5		"	"	"	"		103	72		109	179	104.5	46.5	17.63	18.5	40.25	2542	3086	2217	2326	10121
6					110.22	300	175	57	25	27		"	"	"	"		101	72		97	180	81.0	25.5	11.00	11.75	25.48	1696	1433	1191	1218	5538
7					117.35	285	220	84	32	25.2		"	"	"	"		100	72		104	188	89.5	41.5	12.75	13.75	32.68	1995	2483	1469	1585	7532
8					113.66	274	192	67	28	26.5		"	"	"	"		102	72		100	194	84	28	12	12.5	27.64	1814	1623	1339	1395	6171
9					96.49	295	130	45	20	27.5		"	"	"	"		101	72		92	199	65.	19.5	8.13	8.5	19.67	1191	959	770	805	3725
10					94.85	295	115	40	18	27.5		"	"	"	"		100	72		86	198	59	19	7.0	7.2	17.7	1063	919	652	671	3305
11					95.84	297	123	42	20.5	27.5		"	"	"	"		101	72		90	193	62.7	22.1	7.5	7.81	18.73	1142	1070	712	735	3659
12					77.22	295	75	29	13.8	27.5		"	"	"	"		103	72		84	234	32.5	16.0	4.5	4.6	17.33	550	630	341	349	1870
13					77.67	295	80	32	14.3	27.7		"	"	"	"		105	72		83	236	32.75	18.	4.6	4.25	12.23	476	713	351	324	1864
14					75.52	260	75	27	13.8	27.7		"	"	"	"		108	72		86	193	38	17.5	5.0	5.2	13.2	545	674	371	386	1976
15					61.04	245	47	22	10.6	27.7		"	"	"	"		108	72		81	236	25	13.0	3.25	3.38	9.11	290	405	195	203	1093
16					58.83	260	40	—	10.5	27.7		"	"	"	"		107	72		82	234	22.75	11.88	3.00	3.50	8.55	254	356	173	202	985
17					61.15	260	40	—	10.5	27.7		"	"	"	"		106	72		82	230	22	11.5	2.88	3.50	8.30	256	258	173	210	997
~~Total~~																															
* Average.																															

PORT ENGINE DATA (ONE SHEET FOR EACH ENGINE), STANDARDIZATION TRIALS.

(Fill in with india ink for blueprinting.)

U. S. S. SOUTH CAROLINA, COURSE OFF DELAWARE BREAKWATER, ON AUGUST 24, 1909.

Number of run or observation	Time. H.	M.	S.	Reading of engine counter.	Revolutions per min.	Steam pressure. At engine, per gauge.	H. P. steam chest per gauge.	I. P. receiver abs.	L. P. receiver abs.	Vacuum in inches, mercury.	Barometer.	Cut-off in cyl. in decimal of stroke. H. P.	I. P.	F. L. P.	A. L. P.	Temperature. Outside air.	Engine room working level.	Injection.	Discharge.	Hot well.	Feed, (entering boilers).	Mean effective pressure. H. P. cyl.	I. P. cyl.	F. L. P. cyl.	A. L. P. cyl.	Referred pressure.	Indicated horsepower. H. P. cyl.	I. P. cyl.	F. L. P. cyl.	A. L. P. cyl.	Total.
1					126.94	300	265	110	39	25		80	.72	.44	.44	76	104	74	114	110	210	122.5	45	17	17.3	40.68	2954	2912	2119	2181	10166
2					125.68	280	255	105	38	24.5		"	"	"	"	76	106	72	116	112	195	121	46.5	17	17.8	41.17	2889	2979	2160	2197	10225
3					126.17	300	265	105	40	25.0		"	"	"	"	78	106	74	120	114	193	122	47.5	17.3	17.0	41.30	2924	3058	2144	2107	10230
4					126.60	295	260	105	39	25.0		"	"	"	"	78	106	74	124	112	194	115.5	47.3	17.0	17.3	40.60	2778	3053	2113	2151	10095
5					127.03	295	265	112	41	25.5		"	"	"	"	76	108	72	110	110	192	113.5	48.3	18.1	17.3	41.22	2729	3128	2258	2158	10283
6					113.61	300	215	80	37	26.0		"	"	"	"	76	106	72	116	113	206	96	35.3	12.8	13.0	31.10	2072	2045	1372	1450	6939
7					116.14	290	220	80	35	26.0		"	"	"	"	77	106	72	115	112	200	102.5	37.8	13.1	13.0	32.77	2242	2219	1481	1470	7413
8					113.72	270	205	80	31	26.0		"	"	"	"	78	106	72	114	113	204	107.5	37.8	11.7	12.0	32.05	2322	2192	1307	1340	7161
9					96.02	295	180	70	29	26.0		"	"	"	"	78	106	72	110	114	200	72.8	27	8.0	8.7	22.40	1378	1372	754	820	4324
10					95.51	300	140	55	23	26.0		"	"	"	"	76	104	72	98	100	208	67	24.4	8.0	8.1	20.86	1216	1188	750	760	3914
11					96.02	300	145	52	21	26.5		"	"	"	"	78	104	72	94	98	212	68	25	7.7	8.1	20.96	1240	1224	726	764	3954
12					77.23	295	95	37	15	26.0		"	"	"	"	76	102	72	104	98	196	48.6	19.9	4.5	5.1	14.67	713	784	341	387	2225
13					77.42	280	80	35	14	26.0		"	"	"	"	76	100	72	86	86	206	42.3	18.0	4.7	5.5	13.86	622	710	357	418	2107
14					75.41	265	80	35	14	26.0		"	"	"	"	76	100	72	86	92	240	46.4	18.1	4.7	4.8	13.94	665	696	348	355	2064
15					60.72	255	45	20	9.5	27.0		"	"	"	"	76	100	72	84	86	226	29	13.2	3.1	3.3	9.44	334	409	185	197	1125
16					59.74	265	45	19	9.5	27.0		"	"	"	"	78	110	72	82	86	226	26.5	12.0	2.8	3.0	8.87	301	365	164	176	1006
17 Trial					61.30	270	55	22	10.0	27.0		"	"	"	"	78	110	72	82	86	234	26.5	13.3	3.1	3.2	9.16	309	412	187	193	1105
* Average.																															

PERFORMANCE—FOUR-HOURS' OFFICIAL TRIAL.

Steam pressures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Mean steam pressure at boilers, pounds	285.0	
engines, pounds	275.0	275.0
H.P. steam chest, gauge, lbs	230.0	252.0
1st receiver (absolute), pounds	89.0	104.0
2d receiver (absolute), pounds	32.7	37.4
Vacuum in condensers, inches of mercury, mean	26.88	25.6

Temperatures. (*Average of one-half hourly observations.*)

Injection, degrees	70.0	70.0
Discharge, degrees	111.0	111.0
Feed water, degrees	180.0	199.0
Outside air, degrees	73.5	
Engine room, working platform, degrees	102.0	104.0
Firerooms, working level, degrees	111.5	111.5
Smoke stacks, average, degrees	540 to 560.	
Superheat, main steam pipe, degrees	47.5	
I.P. valve chest, degrees	13.5	(—) 1.4

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

Average revolutions, main engines, per minute	121.27	121.30
Mean revolutions, both engines, per minute	121.28	
Pumps, main air	45.0	45.0
circulating	216.0	198.0
feed, d.s., per minute	46.0	19.0
fire and bilge	22.0	25.0
auxiliary condenser, air and circulating	25.0	25.0
Dynamos	1,700.0	
Blower engines	656.0	
Speed of ship, in knots per hour	18.86	
Slip of propeller, in per cent. of its own speed, based on mean pitch	17.06	17.08
Air pressure in firerooms, in inches of water, mean	0.78	

Cut-off.

H.P. cylinder, in decimals of stroke	0.79	0.80
I.P. cylinder, in decimals of stroke	0.72	0.72
F.L.P. cylinder, in decimals of stroke	0.45	0.44
A.L.P. cylinder, in decimals of stroke	0.44	0.44

Mean Effective Pressures in Cylinders, in pounds per square inch.
(Averages of cards taken at half-hourly periods.)

	Starboard.	*Port.*
Main engines, H.P. cylinder	99.96	109.6
I.P. cylinder	40.37	46.5
F.L.P. cylinder	15.22	15.5
A.L.P. cylinder	15.18	16.1
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	37.04	

Indicated Horsepower.

Main engines, H.P. cylinder	2,302.0	2,526.0
I.P. cylinder	2,495.0	2,877.0
F.L.P. cylinder	1,813.0	1,841.0
A.L.P. cylinder	1,882.0	1,915.0
total	8,492.0	9,159.0
Collective H.P. of both main engines	17,651.0	
Air pumps, main	14.8	14.5
Circulating pumps, main	106.0	95.0
Feed pumps, main	144.0	62.0
Auxiliary condenser, air and circulating pumps	2.0	
Fire and bilge pumps	2.3	3.1
Forced-draft blowers	Electrically driven.	
Ice machines	15.0	
Dynamo engines	242.0	
Total	706.0	
Collective I.H.P., air, circulating and feed pumps	436.0	
Collective main and auxiliary engines in operation	18,357.0	

Coal.

Kind and quality used on trial:
Pocahontas, hand-picked, 14,343 B.T.U. per pound.
Pounds per hour, main and auxiliary engines, during trial 24,620

Deduced Data.

I.H.P. (total) per square foot of grate surface	17.49
Pounds of coal (total) per I.H.P. per hour, collective, main engines, all machinery in operation	1.395
	1.341
square foot of grate surface, per hour	23.47
Cooling surface (main condenser), square feet per I.H.P., main engine	1.076
Heating surface, square feet per I.H.P. (total)	2.29

TWENTY-FOUR-HOUR TRIAL AT 17.5 KNOTS.

This trial began at 4 P. M., August 25, 1909, various courses being steered between Five-fathom Bank Light and Fenwick Island Light. The weather was fair, with light haze clouding up toward end of trial. There were light to stiff breezes from S.W., with moderate sea from same direction.

During this trial all auxiliaries were in operation, including all those usually required under service-cruising conditions, such as dynamos necessary for efficiently lighting the vessel, the ventilating fans operating at their normal speed, the evaporating and distilling plant in continuous operation at its normal rated capacity. The pumping and forced-draft system was operated as required. The following are the data for this trial:

PERFORMANCE.—TWENTY-FOUR-HOURS' OFFICIAL TRIAL AT 17.5 KNOTS.

Steam Pressures. (*Average of hourly observations.*)

	Starboard.	*Port.*
Mean steam pressure at boilers, pounds	252.0	
engines, pounds	236.4	239.52
H.P. steam chest gauge, pounds,	187.4	227.76
1st receiver (absolute), pounds,	67.84	93.38
2d receiver (absolute), pounds,	26.84	33.44
Vacuum in condensers, inches of mercury, mean	28.58	25.96

Temperatures. (*Average of hourly observations.*)

Injection, degrees	71.16	71.24
Discharge, degrees	101.08	101.08
Hotwell, degrees	97.6	100.32
Feed water, degrees	184.76	190.20
Outside air, degrees	74.92	
Engine room, working platform, degrees	97.4	102.44
Firerooms, working level, degrees	110.43	
Smoke stacks, average, degrees	431 to 540	

Revolutions, or double strokes, per minute. (*Average of hourly observations.*)

Average revolutions, main engines, per minute	111.16	110.28
Mean revolutions, both engines, per minute	110.72	
Pumps, main air	60.0	58.0
circulating	102.0	99.0

	Starboard.	*Port.*
Pumps, feed, d.s., per minute	19.8	19.2
fire and bilge	32.8	16.16
Blower engines	675.0	
Speed of ship, in knots per hour	17.68	
Slip of propeller, in per cent. of its own speed, based on mean pitch	15.18	14.49
Air pressure in firerooms, in inches of water, mean	0.6	

Cut-off.

H.P. cylinder, in decimals of stroke	0.79	0.80
I.P. cylinder, in decimals of stroke	0.72	0.72
F.L.P. cylinder, in decimals of stroke	0.45	0.44
A.L.P. cylinder, in decimals of stroke	0.44	0.44

Mean Effective Pressures in Cylinders, in pounds per square inch.

Main engines, H.P. cylinder	83.0	83.75
I.P. cylinder	32.96	38.38
F.L.P. cylinder	11.51	12.44
A.L.P. cylinder	12.09	12.49
Mean equivalent pressure per square inch, referred to combined area of L.P. pistons, pounds	29.515	

Indicated Horsepower.

Main engines, H.P. cylinder	1,774.5	1,779.83
I P. cylinder	1,890.66	2,188.33
F.L.P. cylinder	1,272.0	1,366.5
A.L.P. cylinder	1,336.0	1,371.5
total	6,273.16	6,706.16
Collective H.P. of both main engines	12,979.32	
Air pumps, main	7.1	10.5
Circulating pumps, main	102.0	99.0
Feed pumps, main	60.0	58.0
Auxiliary condenser, air and circulating pumps	0.2	
Fire and bilge pumps	0.2	
Ice machines	15.0	
Forced-draft blowers	Electrically driven.	
Dynamo engines	260.0	
Total	624.0	
Collective I.H.P., air, circulating and feed pumps	336.6	
main and auxiliary engines in operation	13,603.32	

Coal.

Kind and quality used on trial:

Pocahontas, run of mine, 14,547 B.T.U. per pound.

Pounds, per hour, main and auxiliary engines, during trial..... 24,077.0

Deduced Data.

I.H.P. (total) per square foot of grate surface	12.96
Pounds of coal, total, per I.H.P. per hour, collective, main engines	1.885
per I.H.P. per hour, all machinery in operation	1.77
square foot of grate surface, per hour	22.93
Cooling surface (main condenser), square feet per I.H.P. (main engines)	1.46
Heating surface, square feet per I.H.P. (total)	3.47

TWENTY-FOUR-HOUR TRIAL AT 12 KNOTS.

This trial began 4 P. M., August 26, 1909, various courses being run between Fenwick Island Light and Delaware Breakwater. The weather was cloudy and hazy at beginning, clearing up after a few hours. There were gentle to moderate breezes from S.W. to North, with moderate to smooth sea. The data for this trial are as follows:

PERFORMANCE—TWENTY-FOUR-HOURS' OFFICIAL TRIAL AT 12 KNOTS.

Steam Pressures. (*Average of hourly observations.*)

	Starboard.	*Port.*
Mean steam pressure at boilers, pounds	236.0	
engines, pounds	224.0	229.28
H.P. steam chest, gauge, pounds,	91.68	141.4
1st receiver (absolute), pounds	40.94	64.44
2d receiver (absolute), pounds	12.76	17.32
Vacuum in condensers, inches of mercury, mean	27.66	26.03

Temperatures. (*Average of hourly observations.*)

Injection, degrees	71.52	70.8
Discharge, degrees	89.92	89.92
Hotwell, degrees	90.92	95.96
Feed water, degrees	184.12	210.0
Outside air, degrees	76.4	
Engine room, working platform, degrees	97.16	100.2
Firerooms, working level, degrees	107.99	

Revolutions, or double strokes, per minute. (*Average of hourly observations.*)

Average revolutions, main engines, per minute	75.23	75.58
Mean revolutions, both engines, per minute	75.405	
Pumps, main air	18.88	30.12

	Starboard.	*Port.*
Pumps, circulating	180.08	160.2
feed, d.s. per minute	10.67	25.72
fire and bilge	24.72	14.96
auxiliary condenser, air and circulating	17.16	
Dynamos	821.0	
Blower engines	Not running.	
Speed of ship, in knots per hour	12.3	
Slip of propeller, in per cent. of its own speed, based on mean pitch	12.8	13.2
Air pressure in firerooms	Natural draft.	

Cut-off.

H.P. cylinder, in decimals of stroke	0.65	0.69
I.P. cylinder, in decimals of stroke	0.42	0.53
F.L.P. cylinder, in decimals of stroke	0.45	0.44
A.L.P. cylinder, in decimals of stroke	0.45	0.44

Mean Effective Pressures in Cylinders, in pounds per square inch.

Main engines, H.P. cylinder	27.08	32.12
I.P. cylinder	19.29	19.58
F.L.P. cylinder	4.70	4.90
A.L.P. cylinder	4.46	4.59
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	12.21	12.86

Indicated Horsepower.

Main engines, H.P. cylinder	390.33	451.33
I.P. cylinder	743.33	755.0
F.L.P. cylinder	347.67	363.83
A.L.P. cylinder	329.67	339.50
total	1,811.0	1,909.67
Collective H.P. of both main engines	3,720.67	
Air pumps, main	6.5	10.1
Circulating pumps, main	84.0	74.0
Feed pumps, main	37.5	80.0
Fire and bilge pumps	3.5	
Forced-draft blowers	Electrically driven.	
Ice machines	15.0	
Dynamo engines	160.0	
Total	475.6	
Collective I.H.P., air, circulating and feed pumps	292.0	
main and auxiliary engines in operation	4,196.27	

Coal.

Kind and quality used on trial:

Pocahontas, run of mine, 14,312 B.T.U. per pound.

Pounds per hour, main and auxiliary engines, during trial..... 14,312.0

Deduced Data.

I.H.P. (total) per square foot of grate surface.....	5.99
Pounds of coal (total) per I.H.P. per hour, collective, main engines..	2.69
per I.H.P. per hour, all machinery in operation.....	2.385
square foot of grate surface, per hour.....	14.29
Heating surface, square feet per I.H.P. (total).....	6.68

NOTES.

NAVAL AND MILITARY RETROSPECT FOR YEAR 1909.

Progress during the past year in matters naval and military may be considered as decidedly satisfactory. Several ships have been completed, chief among which are the *South Carolina* and *Michigan,* the first all-big-gun battleships of our Navy. These vessels, which are of 16,000 tons displacement, carry each four 45-caliber 12-inch guns. The *Michigan* on her official trial maintained an average speed of 18.97 knots. The *Delaware* and *North Dakota, Dreadnoughts* of 20,000 tons displacement, carrying each ten 12-inch 45-caliber guns and fourteen 5-inch guns, have passed through their trials successfully, the *Delaware,* driven by reciprocating engines, averaging at full power a speed of 21.44 knots, and the *North Dakota,* driven by Curtis turbines, averaging 21.83 knots. The *North Dakota* showed a higher water rate and a lower coal consumption than the *Delaware,* and the naval officials are particularly gratified at the unexpectedly low coal consumption of the turbine-driven ship at cruising speed. Our third pair of *Dreadnoughts,* the *Florida* and the *Utah,* of 21,825 tons displacement and the same armament as the *North Dakota,* will be launched in the spring of the present year, and work is about to be commenced upon those great ships, the *Arkansas* and *Wyoming,* carrying twelve 50-caliber 12-inch guns on a displacement of 26,000 tons. Particularly gratifying has been the speed developed by our latest destroyers of the *Reid* type, the *Flusser* having averaged 32.67 knots and the *Reid* 33.75 knots in five runs over a measured mile. The vessels of this class are fine, seaworthy boats of 700 tons displacement. Equally favorable results have been obtained with our latest submarines, the largest of which, the *Narwhal,* has

maintained a speed of 14 knots on the surface and 10 knots submerged; which performance, so far as we know, has not been surpassed in any foreign navy. During the year the Editor had an opportunity to make a 20-mile trip in one of these vessels, and he can testify to the remarkable ease and accuracy with which the craft was maneuvered. Unquestionably submarine warfare has at last come into its own, and is destined to be a potent influence in deciding the issue of future naval operations. Lattice masts have been fitted to all of our battleships, and, as affording a fire control platform, they have proved a decided success. The gunnery of our Navy continues to maintain its high excellence, and our shooting is believed to be now second to none in the world. Mention should be made of a greatly improved British torpedo, which has a diameter of 21 inches and is credited with a speed of 31 knots over a range of 7,000 yards. The increasing size of battleships has raised the question of increasing the size and draught of drydocks—a most serious consideration. During the year a contract was let for the large drydock at Pearl Harbor, Hawaii, and a new contract has also been let for the big drydock at the Brooklyn navy yard, New York. At the present time only our largest drydocks could float the new *Wyoming* over the sill at high water, and then with but a slight margin to spare.

The new 12-inch 50-caliber type gun, of the kind which is to be mounted in our 26,000-ton ships, has shown, in the proving-ground tests, an initial velocity of 3,030 feet per second and a muzzle energy of 52,500 tons. Greatly exceeding this in power will be the new 14-inch navy gun, recently completed at the Midvale Works, which will fire a 1,400-pound projectile with a muzzle energy of 65,600 tons. The new army 14-inch gun will be less powerful, but its accuracy life will be greater. It will be capable of firing 250 rounds, as against 80 to 100 rounds which is the limit for the present high-velocity 12-inch army gun. A comparison of the sea strength of the powers at the close of the past year places Great Britain first, the United States second, Germany third, France fourth and

Japan fifth. When all ships now building are completed, Germany will be second with 820,692 tons, and the United States third with 789,687 tons displacement. In *Dreadnoughts* Great Britain stands first with seven completed, and nine building; Germany second with two completed, and nine building; and the United States third with two completed, and four under construction. Of pre-*Dreadnought* battleships carrying guns of 11-inch caliber or over, Great Britain has forty-nine; the United States twenty-five, and Germany fourteen.

MERCHANT MARINE.

The deplorable decadence of our merchant marine has continued throughout the year, and we look in vain for any adequate evidence of the awakening of the nation to the seriousness of this pre-eminently national question. As a measure of security and defense, the existence of an adequate number of merchant ships to serve as transports and colliers in time of war is vital to the efficiency of our Navy. Although, during the world cruise of our fleet the world looked on approvingly and applauded this evidence of material strength, the people who know—the naval boards of strategy and naval officers in general—must have smiled as they realized that, because of our want of transports and colliers, a voyage of this character would never be seriously contemplated by our naval board, and certainly never be attempted. The signs of decadence of our merchant marine are so clearly written that he who runs may read. Within two years the number of American steamers crossing the Pacific and capable of carrying the mails has been reduced more than one-half. The year before last the Post Office Department recommended and the Senate passed a bill providing for a compensation of so much per mile to steamers running to South America, the Philippines, Japan, China and Australia; but the measure failed to become law. It is certain that without such federal encouragement American steamship lines will never be established, and until the con-

struction of mail steamers and freighters is encouraged, our splendid Navy will be robbed of its efficiency and limited, at least in the opening months of a war, to the defense of its own ports.

The past year will be notable in the annals of transatlantic travel for the fact that a transatlantic liner made the passage for the first time at an average speed of 26 knots an hour. This was accomplished last October, when the *Mauretania* covered the westward course from land to land in 4 days, 10 hours, and 51 minutes, at an average speed of 26.06 knots. Both this vessel and her sister, the *Lusitania,* are now capable of 25 knots sustained speed in average weather and 25½ knots during the quiet weather of the summer season. Work on the two enormous White Star boats, the *Titanic* and *Olympic,* has proceeded rapidly. The former vessel is already in frame, and she will probably make her first trip in 1911. The dimensions of these ships, as announced by the company, are: 790 feet over all, 92 feet beam, and maximum displacement 60,000 tons. The speed is to be 21 knots with 45,000 horsepower. The success of the marine turbine has been settled beyond all question. The substitution of four-bladed propellers of smaller diameter for three-bladed propellers on the outer shafts of the Cunarders has not only completely eliminated what vibration there was, but by improving propeller efficiency has considerably increased the speed. There is no evidence that any company will attempt to rival these vessels in speed, and probably future development will be along the lines of the *Olympic* and *Titanic.* These moderate-speed vessels are to be driven by a combination of reciprocating engines and turbines, the reciprocating element being used in the higher ranges of expansion, in which it is more economical than the turbine. In a recent trip to New Zealand a merchant vessel, the *Otaki,* fitted with engines and turbines, made the same average speed as the sister ships *Orari* and *Opawa,* fitted with reciprocating engines alone. Her coal consumption was 11 per cent. less, and there was a reduction of 20 per cent. in the water consumption, all three ships having the same boiler installation.

In this combination, when reversing, the turbine is cut out and the reciprocating engines are connected directly to the condenser. Toward the close of the year two interesting devices, designed to reconcile the slow-speed demands of the propeller with the high-speed demands of the turbine, were made public. One, designed by Admiral Melville and Mr. Macalpine, consists of a reduction gear of the helical type interposed between the turbine and propeller shaft. The other, designed by a German engineer, employs a form of hydraulic turbine transmission, in which the ratio of turbine speed to propeller speed can be varied indefinitely. For both devices a high efficiency rate is claimed. The Curtis turbine, because of its larger diameter, and comparatively low speed of revolution, has less trouble from propeller inefficiency than the Parsons type. The loss of the *Republic* early in the year gave dramatic evidence of the value of wireless telegraphy as a safeguard to the safety of passengers and ships. It served also to draw attention to the question of bulkhead protection. Theoretically, the *Republic* should have stayed afloat long enough to be brought into port. It is probable that she sank because of the gradual failure of her bulkheads, one by one. On the other hand, the value of bulkhead protection was proved by the fact that the colliding ship, the *Florida*, with 30 feet of her bow crushed in, was able to make port in safety.

An important event for the navigator was the launch of the magnetic survey yacht *Carnegie*, into the construction of which no steel, iron, or other magnetic material entered. She was built for the Carnegie Institution at Washington, and her surveys of the ocean will form part of a comprehensive survey of the whole world on land and sea for the determination of the exact local magnetic variations of the compass. An interesting novelty in hull construction was seen in the *Monitoria*, whose hull is built with large corrugations, the object of which is to increase the longitudinal strength of the ship, without increasing the weight. The extra cost is slight, and the carrying capacity is said to be increased from three to four per cent.—"Scientific American."

MARINE ENGINEERING.

In marine engineering interest centers in the turbine. For some inexplicable reason it is almost impossible to overcome the secrecy observed as to its performance by those who use it; and the not unnatural deduction is that there is a good deal to conceal. The facts leak out, however, and the experience acquired is that, so far as economy of fuel measured by displacement ton-mile is concerned, the turbine is not more economical than a good compound engine. Estimated in terms of horsepower, it is very little, if at all, more economical than a good triple-expansion engine. The most accurate, complete, and conclusive evidence available is furnished by a report on the performance of the United States warships *Delaware* and *North Dakota,* published in our impression for December 17th. Making allowance for the difference in shaft horsepower and indicated horsepower, the figures show that a small economical advantage lay with the piston engines. Particular interest lies in the circumstance that the losses seem to be nearly the same in amount in both engines. Now the combined diagrams of the *Delaware's* engines show abnormally heavy losses in the shape of "missing quantity." The gaps again are very large; so are clearance spaces. We shall not be far wrong if we take the missing quantity as amounting to 30 per cent. The turbine is not supposed to suffer any loss of steam by internal condensation. Its range of expansion is very much higher than that of the piston engine. The back pressure is as nothing. It is always maintained that if back pressure and cylinder condensation could be reduced or eliminated altogether, an enormous economical advantage would be gained. In the turbine these conditions are nearly satisfied, and yet the turbine can hardly hold its own with its older rival. Be the explanation what it may, it is certainly a curious coincidence that the thermal efficiency of the two types should be so nearly alike.

Propeller Efficiency.—The driving efficiency of the marine engine is inseparably bound up with that of the propeller. The

theory of the performance of propellers has been energetically discussed in our columns during the past year. Without going into side issues, it may be taken as settled that slip represents the principal loss of efficiency. The thrust of the propeller on the water one way is precisely equal to that on the ship the other way. Each represents a moving effort; and the efforts being the same, the distances passed over in a given time represent the powers expended. If, for example, the velocity of the thrust on the water pushing it astern is 10 per cent. greater than the velocity of the thrust pushing the ship forward, then the loss is 10 per cent.; and the statement does not contradict Rankine's proposition that the greater the weight of water pushed astern and the less its velocity in a given time, the higher will be the efficiency, because this, put into other words, simply means that these conditions secure the least slip. There is another source of loss quite independent of slip, namely, blade friction in the water, and the production of eddies. These losses seem to increase in a very rapid ratio with increased rapidity of revolution and reduction in size of the screw. It may be taken for granted that the slip in the screws of such vessels as the *Mauretania* and *Lusitania* must be very large, and this is confirmed by the circumstance that four blades have been and are being tried with apparently considerable benefit, instead of three blades. Sir W. White has recently given some information concerning the performances of these two gigantic ships in the columns of the "Times." He has, however, only said enough to make us wish for more. As to consumption of fuel, all that he can tell us is that which was already made known by Mr. Bell, namely, that on her third westerly voyage the *Lusitania* burned 5,000 tons of coal between Queenstown and New York. For the rest, we reproduce Sir William White's own words: "Naturally the Cunard Company does not feel bound to publish the results of its subsequent experience and the present rates of coal consumption. Some critics have inferred that this official silence has been maintained because the results have been unsatisfactory; as a matter of fact, it may be stated without breach of con-

fidence that very substantial economies have been obtained, and that the early results given by Mr. Bell, excellent as they were for a beginning, do not represent present practice." The only comment we have to make is that the next great Cunard fast liner is to have reciprocating engines. The *Mauretania* has so far made the most rapid passages. The highest average speed attained by her on a westward passage has been 26.06 knots (end of September), and the highest average speed going eastward has been 25.89 knots (August). On five consecutive westward passages (July–September) the average speed was 25.7 knots, and for five consecutive eastward passages (May–August) the mean speed was 25.75 knots. We have no information whatever as to the power exerted during these voyages.

The Upkeep of Turbines.—Apart from the question of economy of fuel, we have to consider the general cost of upkeep—a point about which little or nothing has been published. A very interesting article on the upkeep of the Parsons turbines of the United States ship *Chester,* written by Lieutenant Yates, United States Navy, will be found in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS for November. A good deal of trouble seems to have occurred with the astern turbine, apparently due to the presence of condensed water, which stripped the blades.

The Necessity for Perfect Condensers.—It is known as a scientific bit of information that a high vacuum is essential to the efficient performance of the turbine. Consequently the design of the condenser and air pump is a vital factor. But it is not so generally known that the condenser must be free from water leaks to an extent not at all necessary with piston engines. A paper read before the Liverpool Enginering Society by Mr. C. J. Blackburn throws a good deal of light on the point. He says: "The evils arising from a leaky condenser are well known to those who have had to do with turbine machinery. When the condensers are tight and the boilers are producing a good supply of dry steam, all goes well with the turbine, but what a change takes place when the

condensers leak badly! For economical working dry steam is essential, and its presence is made manifest to the engineer by the whistling sound produced in its passage to the turbine. But it is good-bye to dry steam if the condenser leaks, and if the leaky condition is allowed to continue not only is the economy affected, but the very life of the turbine may be materially curtailed. Impurities are carried into it which to some extent block the small openings between the blades, roughen the smooth surfaces, and set up corrosion of the dummy rings, the interior of rotors, and the casings." This testimony is far from standing alone. The marine turbine is in one way exceedingly simple, but it demands all the refinements of construction in itself and its accessories hitherto peculiar to mathematical instruments. It has been well said that it bears the same relation to the normal marine engine that a ship's chronometer bears to a "grandfather's" clock. It is fortunate, too, that it does not require much in the way of repairs, for up to the present accessibility has not been secured. Even to lift the top of a turbine casing is not an easy matter, as the weight is very considerable and there are large joints to be broken. When the top is fairly up the rotor can be examined by turning it round, but to get at the bottom casing means the lifting of the whole rotor, an operation not to be lightly undertaken when we bear in mind that the clearance is so small that the least tipping of the shaft is sure to bend or break many of the blades.

A Curious Boiler Trouble.—As regards smaller steamers of the Channel class, we understand that an unaccountable trouble has arisen with the boilers, which demand quite unusual care to prevent corrosion. The zinc plates disappear rapidly, and it is becoming the practice to introduce zinc powder with the feed water.

Reciprocating Engines and Turbines.—About the combination of the turbine with the reciprocating engine—"the mixed grill"—we are unable to supply very much information, because the results obtained with such ships as the *Laurentic* are kept quite secret. An interesting paper was, however, read

before the joint meeting at Glasgow of the Institution of Engineers and Shipbuilders in Scotland and the North-East Coast Institution of Engineers and Shipbuilders by Engineer-Commander W. McK. Wisnom, R. N., on this subject. The paper dealt with comparative results obtained with the New Zealand Shipping Company's vessel the *Otaki,* fitted with a combination of turbines and reciprocating engines, and with the same line's boat the *Orari,* which only has reciprocating engines, both these ships having been built by Denny's, of Dumbarton. According to this paper the running of these vessels—as far as it had gone—showed a saving in fuel, as regarded the *Otaki,* of some 12 to 15 per cent. Several vessels are being fitted with the combination, but so far as we can learn each new ship must be regarded as an experiment. We must once more call attention to the fact that this is purely a shipowner's question, and economy of fuel is only one out of several factors going to settle what type of propelling machinery is the best for a ship run on ordinary commercial lines.

High Speed Turbine Ships.—The turbine has so far been found suitable only for fast steamers, and its greatest successes have been in Channel passenger boats, among which special mention may be made of the *Ben-my-Chree,* plying between Liverpool and Douglas, in the Isle of Man. This fine ship is 375 feet by 46 feet, with a displacement of 3,350 tons. Her shaft horsepower is 14,700. Her average speed is about 23.88 knots in the open sea, the port low-pressure propeller making 461.5 revolutions, the high-pressure propeller 453.2, and the starboard low-pressure propeller 458.8 revolutions; boiler pressure 166 pounds, vacuum 27 inches. It seems to be admitted that the coal bill is heavy; but, on the other hand, there appears to be a considerable reduction in the cost of engineers, lubrication, packing, and so on.

Proposed New Methods of Driving.—The popularity of the turbine with the traveling public is of considerable moment—worth having, indeed, although something has to be paid for it. It is estimated that at the end of last year there

were 275 vessels fitted with Parsons turbines afloat, representing over three millions of shaft horsepower. It is but natural that attempts should be made to get better results from the combination of the turbine and the screw propeller than have hitherto been obtained. These all take the same direction. The aim is to run the propeller slowly, while the turbine runs fast. The first scheme is simply to let the turbine drive a dynamo, while the dynamo drives a motor on the propeller shaft. Much can be said for this idea and much against it. The second and less plausible idea is to let the turbine drive high-tension—series—centrifugal pumps, the water from which shall drive a water turbine on the propeller shaft. The last scheme originated, we believe, with Admiral Melville, a veteran naval American engineer, and is a reversion to gearing. When screw propulsion began the engines ran too slowly for the propellers, and gear was introduced to run them quicker. Admiral Melville has devised a very ingenious system of helical gearing, and to avoid noise and backlash he introduces what is termed a "floating ring," which is to enable the small wheel on the turbine shaft to keep always in strict alignment with, and the proper depth in, the spur wheel on the propeller shaft. This is necessary because the width of the wheel and pinion may be as great as several feet. When we find that the gearing must be adjusted to at least the one-hundredth part of an inch, and that silence depends on the prevention of actual contact between the teeth by a film of oil, it will be seen that the mechanism has yet to prove its suitability to the somewhat rough-and-ready conditions prevailing in a marine engine room. A good deal of attention has been bestowed during the past year on condensers, but there is really next to nothing new to record of marine-engine work, nor are there any new departures probable in the present year.

Four-Crank Engines.—The tendency is to increase the number of cranks, and, of course, of cylinders in the cargo as well as in the passenger-carrying boats of the mercantile marine. Engines with four cranks are now quite common. At

the time, a few years ago, when the late Thomas Mudd introduced engines with five cranks it was held that he was, to say the least, unwise; but experience has gone far to justify his beliefs. Every day it becomes more certain that the mere multiplication of stages of expansion will not secure economy. The missing quantity is as large a percentage in the triple and quadruple-expansion engine as it is in the two-cylinder compound. As we have repeatedly pointed out, the superior economy of the multiple-expansion engine is due to the use of high pressure and long ranges of expansion, not to the equalization of cylinder temperature. This fact has led to a progressive increase in boiler pressures, and 200 pounds and even over, is now in daily use. Such pressures are best dealt with by subdividing the power of the engine among a number of cylinders. With the four-crank engine the turning moment is greatly improved, the stress on crank and propeller shafts diminished, balancing is simplified, and vibration annulled. It is maintained also that the four-crank engine is less likely to race, and that the evil effects of racing are diminished. As to cost, there is no reason to believe that a triple-expansion engine with four cylinders is much more expensive than one with three, because the weight of individual castings is reduced. Be that as it may, it is clear that, in the opinion of the shipowner, the advantage gained balances the augmented first cost.

Superheating.—Little or nothing is being done in the way of superheating at sea. That it would reduce the consumption of steam per horsepower per hour is admitted, but the practical difficulties in the way of its introduction are very great. Although friction does not come in, it is dangerous to use highly superheated steam with turbines, because it is apt to cause distortion, which the clearance is much too small to tolerate. It is an easy matter to dry steam without raising its temperature far, because moist steam will take up heat very freely. But to heat gaseous steam is a troublesome undertaking. It is necessary that the temperature of the superheater pipes shall be very high. In a well-designed Scotch boiler, big

enough for its work and not forced, the smoke-box temperature will not much exceed 700 degrees. At least twice this is required to raise steam to 550 degrees or 600 degrees. In locomotives and on land this can be had on the Schmidt system and its congeners by putting in certain large flue tubes, which are filled with flame when the blast pipe pulls hard. The Scotch boiler does not lend itself to this arrangement; and, whatever be the cause, it has been abundantly proved that at sea superheater tubes are extremely short lived.

In the mercantile marine the Scotch boiler is, if possible, more popular than ever. In the Navy there are only three types worth naming, the Yarrow, Babcock, and specially designed Scotch boilers. There is not the least indication that anything new in the way of steam generators will find adoption during the present year.—"The Engineer."

SEA STRENGTH OF PRINCIPAL NAVAL POWERS.

Toward the close of each year our Navy Department, through its Office of Naval Intelligence, issues a comparative table showing the warship tonnage of the principal naval powers. It is based on the number and displacement of warships built and building of 1,000 or more tons, and of torpedo craft of more than 50 tons. The statement for the present year, which shows the relative standing of the navies on November 1st, possesses special interest because it gives an authoritative statement as to the present strength of the navies in ships of the *Dreadnought* type—a subject regarding which the public has been treated during the past few months to overmuch literature of a sensational and misleading character.

Of battleships of the *Dreadnought* type Great Britain has afloat and completed four, Germany two, and the United States two. France, Japan, Russia, Italy and Austria have not as yet completed a ship of the *Dreadnought* type. Of *Dreadnoughts* under construction, Great Britain possesses

seven, Germany six, the United States four, Japan two, Russia four, and Italy one. France and Austria have no *Dreadnoughts* under construction. In this connection it should be mentioned that our Navy Department does not consider that a battleship is entitled to be reckoned as of the *Dreadnought* type unless her main battery consists entirely of guns 11 inches or more in caliber. This eliminates the six French battleships of the *Danton* type, carrying four 12's and twelve 9.4's, and the three Austrian battleships of the *Ferdinand* type, carrying four 12's and eight 9.4's.

Of armored cruisers of the *Invincible* type, Great Britain possesses three, and has two under construction; Germany has none completed, and three under construction; and Japan has one completed and one building. Adding together the totals for ships armed entirely with big guns, both *Dreadnoughts* and *Invincibles,* we find that Great Britain has seven completed of 125,450 tons displacement, and nine under construction of 191,000 tons; Germany has two completed of 36,000 tons, and nine under construction of 183,000 tons; the United States has two completed of 32,000 tons, and four under construction of 83,460 tons; Japan has built one of 14,600 tons and is building three of 56,200 tons total displacement; Russia has none completed and is building four of 92,000 tons total displacement; Italy has none completed, and one of 18,600 tons under construction; France and Austria have nothing either afloat or on the stocks of the all-big-gun type.

Of battleships of the first class, other than *Dreadnoughts* (in which enumeration the Navy Department includes all battleships of about 10,000 tons displacement or over that are less than twenty years old, unless they have been reconstructed and re-armed since 1900) Great Britain possesses forty-nine of 714,750 tons displacement; Germany, twenty-four of 282,424 tons; the United States, twenty-five of 334,146 tons; France, seventeen of 215,270 tons; Japan, twelve of 171,898 tons; Russia, five of 166,000 tons; Italy, ten of 122,600 tons; and Austria, three of 31,800 tons. Great

Britain, Germany, the United States and Italy are building no battleships of this class, but France has six, Japan one, Russia four, and Austria three under construction.

Of armored cruisers other than the *Invincible* type, Great Britain possesses thirty-five of 416,600 tons displacement; Germany, nine of 86,693 tons; the United States, twelve of 157,445 tons; France, twenty-one of 192,982 tons; Japan, eleven of 180,900 tons; Russia, seven of 70,200 tons; Italy, eight of 59,000 tons; and Austria, three of 18,800 tons.

In the class of torpedo-boat destroyers, Great Britain leads with one hundred and forty-eight, followed by Russia with ninety-seven, Germany with seventy-nine, Japan with fifty-six, France with fifty-six, the United States with seventeen, Italy with seventeen, and Austria with six. Great Britain has twenty torpedo-boat destroyers under construction, the United States nineteen, Germany eighteen, France sixteen, Japan three and Austria six.

The following table gives the relative order of warship tonnage both at present and when all the ships now under construction are completed. It will be noted that the only change in relative standing in the second list is that Germany will take the second position at present held by the United States:

RELATIVE ORDER OF WARSHIP TONNAGE.

Nation.	At present, tonnage.	With all vessels completed, tonnage.
Great Britain	1,758,350	2,005,873
United States	682,785	785,687
Germany	609,700	820,692
France	602,920	766,906
Japan	396,368	489,704
Russia	259,263	412,250
Italy	216,038	257,818
Austria	114,897	167,297

Referring to the statistics of total displacement, particularly of battleships of the pre-*Dreadnought* type, we wish to draw attention to the fact that although Germany possesses only

one battleship less than the United States, the average displacement of the German ships is very much smaller, being about 11,770 tons as against 13,370 for the United States battleships. Furthermore, our strength in this class of vessel is incomparably stronger, since every one of these twenty-five ships carries a main battery of four guns of 12 or 13-inch caliber, whereas ten of the German battleships carry nothing heavier than a 9.4-inch gun, a weapon which at the great ranges of 7,000 to 9,000 yards at which modern battles will be fought would be altogether ineffective against battleship armor. Hence, as late as a year ago, when Germany had no *Dreadnoughts* afloat, she actually possessed only fourteen battleships capable of fighting effectively at modern ranges, as against twenty-five flying the United States flag, and forty-nine under that of Great Britain.

It is here that the careful observer of naval development must look for an explanation of the feverish haste with which Germany is building a fleet of *Dreadnoughts*. The short-sighted policy which led to the mounting of the 9.4-inch gun as the principal arm in her first two squadrons of battleships, left her Navy in a very serious condition when the absolute superiority of the 12-inch gun was demonstrated in the battle of the Sea of Japan. Her present lavish expenditure on battleships must be regarded rather in the light of an effort to retrieve a disastrous error than as a distinct challenge to the supremacy of the British or any other Navy.

Nor is it any answer to this view of the case to state, as has so often been done of late, that the construction of the first *Dreadnought* robbed all existing battleships of their right to be named as such. We confidently predict that the battleships of the older class will play an unexpectedly important rôle in the next great war; and when two theoretically unsinkable fleets of *Dreadnoughts* have hammered each other into a state of comparative exhaustion, it will be the nation that can send in the most numerous second line of older ships, armed with a mixed battery of 12's and smaller rapid-fire pieces, that will win the day.—"Scientific American."

THE NEW FOURTEEN-INCH ARMY AND NAVY GUNS.

Great interest attaches to the new 14-inch gun for the Navy which has been delivered at the navy yard at Washington, where it will be rifled and the breech mechanism installed. On completion it will be shipped to the naval proving grounds at Indian Head, Maryland, for a test. The completed gun will weigh about 63 tons; its projectile will weigh about 1,400 pounds, and the latter will be propelled by a charge of 365 pounds of powder. The extreme range at maximum elevation will be more than twenty-five miles. As compared with the present navy 12-inch gun, this will be a far more powerful weapon. It will penetrate any armor afloat, at the most distant ranges at which accurate aiming is possible. The army has also a 14-inch gun of much lower velocity and power, which will soon be tested at the Sandy Hook proving grounds. The lower muzzle velocity (about 2,000 feet per second) of the 14-inch gun will give it additional life over the 12-inch gun. After some eighty discharges the rifling of the present 12-inch gun of 2,500 foot-seconds velocity becomes so badly worn as to destroy the accuracy. In the case of the 14-inch gun the erosion is much less, and the gun will be serviceable for about 300 discharges.

THE NEW BRITISH AND GERMAN "DREADNOUGHTS."

The biggest *Dreadnought* of Great Britain and the most powerful German vessel of the same type were put into the water on the same day. The *Neptune,* the English vessel, has a displacement of 20,250 tons, will carry ten 12-inch guns and twenty 4-inch guns, and her total broadside will weigh 9,120 pounds. The German *Dreadnought, Ostfriesland,* has a displacement of 19,000 tons; will carry twelve 12-inch guns, twelve 6.7-inch guns, and twenty-two 4-inch guns, and she will deliver 14,204 pounds in a single discharge of all her guns. The most remarkable feature of the *Neptune* will be

the extraordinary power of concentrating the gunfire. The ten 12-inch guns will be mounted in pairs in five barbettes. One barbette will be placed in the forecastle on a high level. The two broadside barbettes will be *en echélon,* that on the port side being more forward than the barbette on the starboard side. By a novel arrangement the superstructure is bridged over these two barbettes, both pairs of guns being fired on either side at once if required. Of the two barbettes astern, one will be raised above the other, so that all four guns can be fired direct astern simultaneously. The broadside fire of the *Neptune* will thus be ten guns, the stern fire eight guns, and the direct-ahead fire six guns. The *Ostfriesland,* on the other hand, has four of her twelve guns masked by the superstructure, and can fire only eight 12-inch guns on either broadside. Her end-on fire will be six guns ahead and astern.—"Scientific American."

THE GERMAN NAVAL ARCHITECTS.

The eleventh regular meeting of the Schiffbautechnische Gesellschaft was opened on the 18th inst. in the Grand Hall of the Königl Technische Hochschule in Charlottenburg by the president of the Institution, the Grand Duke of Oldenburg, who, after reading a telegram from the Emperor expressing regret that he was unable to be present, invited Professor Stumpf to read his paper on "Continuous-Current Steam Engines."

The continuous-current steam engine, said Professor Stumpf, is a development of the ordinary steam engine in the direction that the steam behind the piston exhausts at the end of the stroke through a row of ports at halflength of the cylinder. The length of the cylinder is about twice that of the piston itself plus the length dimension of the ports, which, by the passage of the latter, are laid open to the steam space. The steam thus does work till the completion of the stroke

and has not to return to the end of the cylinder to make its escape. The cooling of the cylinder walls, and consequent condensation, are thus reduced to such an extent that successive grades of expansion become unnecessary, and a return can be made to the single-cylinder engine. The exhaust ports have about three times the area of ordinary valves, which practically increases the efficiency of the condenser. The piston itself taking the place of the ordinary exhaust valve, the losses due to steam, leakage, friction and extra complication are avoided. The admission ports are in the cylinder covers, which are thus warmed at the outset. During the stroke the hottest steam is that at the cylinder end and the coolest and dampest at the piston; the steam, then, with its condensation losses, is carried off at the end of the stroke through the ring of ports. Exhaust steam in no case passes over a heated surface in the cylinder, and there are thus, it is claimed, none of the familiar heat losses. At the ends of the cylinder there is a certain amount of space in which the remaining steam is compressed, its temperature being thereby raised. Eperiments are to be made to determine the best amount of clearance to be thus given.

The continuous-current cylinder bears somewhat the character of two ordinary single-acting cylinders joined together at their exhaust ends. The temperature changes gradually from the admission to the exhaust, and the diagrams obtained are good. Experience shows that high temperatures may be resorted to without the slightest difficulty, and with good work the piston packing gives excellent results. Further, the replacement of a piston is easily and cheaply effected. The new system admits of the employment of much higher temperatures than usual, and in this direction promise is given of new developments. The arrangement of the admission valves in the cylinder cover shortens the passage of the steam into the cylinder to the greatest possible extent, and this circumstance also favors the use of steam of very high temperature. The consumption of steam is as low as in the cases of compound and three-cylinder engines of the ordinary pattern.

With regard to applications, it is shown that the system lends itself well to atmospheric exhaust, condensing and superheated steam engines, the advantages being especially apparent in connection with the two latter systems. The stresses on the driving gear are unusually small, and the gear can therefore be of relatively small dimensions, while in the mountings and connections simplifications are possible in various ways. Illustrations are given of several installations of this kind of powers ranging up to about 500. A comparison made between a continuous-current engine and one with triple expansion, both in actual work, shows that the number of journals, pistons, pairs of wheels, stuffing boxes and eccentrics amounts in the first case to 33 and in the second to 228, the consumption of oil being somewhat in the same proportion.

The advantages of the system are further shown as applied: (1) To steam locomotives, in which the coal consumption is about 20 to 30 per cent. greater in the ordinary engine than in the new type. (2) To portables, in which the combination of boiler, engine, etc., in one is very much facilitated, and the simplification of the parts is such as greatly to favor wholesale manufacture. (3) To rolling-mill machinery, which, with increased efficiency, is considerably simplified. (4) To winding engines, which are estimated to cost one-third less than those of the older patterns. (5) To compressors, blowing machinery and pumps. (6) To the propulsion of ships, in which the power can be divided over several single cylinders, which, in cases of breakdowns, can each work independently. In this case also steam at high temperature can be taken from the main engines for many purposes for which auxiliary engines have hitherto been provided. Here also the loads on large-powered machinery are stated to be less than those in ordinary engines. In conclusion, Professor Stumpf sets forth that the expansion grades represented by the compound and three-cylinder engines are superfluous complications, and, as a proof that the continuous-current system has passed the experimental stage, gives a long list of manufacturers who have adopted or are adopting it.

The second paper was entitled "Eine neue Lösung des Schiffsturbinen Problems." (A new solution of the marine-turbine problem.) It was written by Dr. Ing. Föttinger, Stettin, and a *précis* is given below.

The design of the hydro-dynamic transformer, which is the subject of this paper, was worked out by Dr. Föttinger, and about the close of 1896 laid before the managers of the Stettiner Maschinenbau A.-G. Vulcan, who at once took the matter up, and after obtaining the necessary patent protection, put in hand a design for a 100-horsepower engine on the lines proposed. The well-known objections to the universal adoption of the turbine for ship work are: The wide differences in speed suitable for turbine and propeller, respectively, the difficulty in reversing with the arrangements hitherto adopted, and the relatively large space and weight required for low-speed turbine plants.

A compromise having to be made between the speeds of turbine and propeller, the former loses about 10 to 15 per cent., and the latter about 30 per cent., so that the net result of the combination is 55 to 60 per cent. of what might be achieved if each of these parts could work under conditions favorable to its own efficiency. Reference is made to the mechanical gearing of De Laval, and to the attempts now being made in America by Melville and others to apply this mechanism on a large scale. The difficulties here met with are pointed out, and a fatal objection to such systems is considered to be the noise made by the toothed wheels. The electric-power transmission also having hitherto proved impractical for marine work, by reason of the great increase of weight of machinery occasioned by it, Dr. Föttinger turned his attention to water power, and he has now produced a hydro-dynamic turbine, the gradual development of which is described.

The fundamental idea is that of a turbine driven by a shaft and drawing water from a cistern. This water is projected through a pipe and nozzle on to the blades of another turbine, which is thus made to turn a second shaft, the water meanwhile exhausting back into the original receptacle, so that a

continuous fresh supply is unnecessary. In the new apparatus these two turbines are combined in one case. Two shafts—I and II—set in a line meet together within the case. The case is concentric with the shafts. A longitudinal section through the apparatus discloses a system of turbine wheels taking an oval from above and below the shafting. If we consider the oval above the shafting, it is seen to be divided into a right-hand semi-oval, C, and two left-hand quarter ovals, A and B, the semi-oval being a fixture in one with the case, and the quarter ovals, each containing a revolving wheel. All three parts are bladed. The turbine wheel A in the inner quarter oval on shaft I imparts the energy given by the latter to some fluid such as water, partly by acceleration and partly by increase of the hydraulic pressure. The jets issuing from A are led directly on to the wheel B—that is, in the outer quarter oval, and connected with shaft II. From B the water passes into C, from which it is led back again to the pump. The process then begins again. The arrangement of the parts A, B and C can be varied in different ways. In another arrangement illustrated, B is fixed instead of C, and the two shafts turn in different directions. This arrangement is suitable for reversing. According to degree of gearing down required, the system of turbines can be doubled or trebled as desired. An arrangement is illustrated in which two oval turbine systems of the kind described are placed side by side in a single case.

These arrangements are notable for their extreme simplicity, and the loss of power in the apparatus is very small. With a six-fold speed reduction obtained by a double transformer, an efficiency of 80 to 82 per cent. is obtained, while eight-fold gearing requires a treble apparatus, and gives about 80 per cent. efficiency. By a suitable utilization of the heat imparted to the water in the process, it is estimated that a saving of from three to four per cent. may be attained, so that the real efficiency is increased by this amount.

To facilitate reversing, the apparatus is further developed, so that the fixed part may have bladings differing in direction,

which may be thrown in or out of gear by the turning of a handle. Such regulation of the apparatus as may be required can be effected by a similar manipulation of the fixed bladings or by other means.

In 1907 the Vulcan Company built an experimental apparatus of the double type, which was tested in every imaginable way. The results were so good that a small vessel was built to test the apparatus, the power of which was raised from 100 to 500 by an increase in the speed of revolution. The gearing down was about 5.6 to 1. The boat was tried under all possible conditions with complete success. In going astern the efficiency was as high as 85 per cent. In the course of these experiments and of the working out of the many designs based on their results, it appeared that the apparatus possessed the following advantages: (1) Propellers of the large standard patterns may be used with high-speed turbines. (2) The reversing conditions are vastly improved. (3) In view of the increase of efficiency attained the main turbines could be reduced in size. (4) The maximum steam pressures in the main turbines may be reduced considerably, and the cases can therefore be lightened or made of cast iron instead of the stronger materials now necessary. (5) The danger of shafts breaking is considerably lessened. (6) The weight of the machinery and the space occupied by it are very much reduced.

The third paper was entitled "Schwere Werftkrane für die Schiffsausrüstung" (Heavy Shipyard Cranes for Fitting-out Purposes), and was written by C. Michenfelder, of Düsseldorf.

This paper is practically a continuation of another one delivered last year on the crane equipment of building ships. Attention was this time given to the second stage of ship construction—the shipping of engines and boilers and other heavy weights—for which larger and larger cranes have yearly to be provided. A great number of illustrations were given, and the advantages and disadvantages of the different systems in use in Russia, Japan, England and Germany pointed out the costs of installation and working of these giants, and their permanent advantages in facilitating and cheapening the oper-

ations of the shipyard, were dwelt upon. Finally, Herr Michenfelder gave expression to the hope that closer and more cordial relations might be brought about between shipbuilders and crane builders, and that a further development of shipyard lifting appliances might be brought about.

"Fabrikorganisation, besonders der Werften." (Organization of Works—in particular that of shipyards.) By Herr Gümbel, of Bremen, was the fourth paper considered.

Herr Gümbel, who, under the Hamburg-Amerika Line, at one time had special opportunities of studying the organization of many widely differing German and English shipyards, gave his ideas on the bookkeeping and general commercial arrangement of an ideal large concern, comprising shipyard, engine works, foundries, etc. He contended that the bookkeeping should be kept distinct for each of the departments, so that the expenses of all kinds attaching to each of these and, as far as possible, those in connection with each article manufactured could be located and a general review of the real conditions in every case rendered possible. Departments devoted to operations similar in their nature should, he considered, be placed under the same head. The arrangement of piecework rates and such like duties would by his system be taken from the foremen and given to a body of clerks and timekeepers, who would also relieve the managing staff of all commercial correspondence. Better co-operation should be obtained between the different branches by improvements in the telephonic and other means of communication, as well as by daily conferences between the departmental managers. Some space is given to the importance of storekeeping, the principle being laid down that, while the stores themselves should be kept by the various departments, their administration should be centralized. The store department should attend to all buying and selling of materials and to the dispatch of finished articles. After touching on the necessity existing for hand-in-hand working between the designing and the executive departments, Herr Gümbel entered upon a general discussion of the management, and of the methods to be employed in the

commercial departments, and of the utilization of the statistical information thereby obtained in the fixing of wages, selling prices, etc., and in determining the real factors influencing profit and loss.—"The Engineer."

TURBINES IN THE UNITED STATES NAVY.

It is impossible to over-estimate the naval importance of an experiment which has been very recently carried out by the United States Government. It is a matter of almost common knowledge that the efficiency of a warship depends largely on her range of action. In other words, how far she can go without having to refill her bunkers. This depends on certain factors, which are all variables. Thus, let us suppose that the consumption of fuel varies as the square of the speed. Then doubling the speed augments the consumption per hour or per day fourfold; but for any given distance the consumption is only doubled because the number of days or hours of steaming is halved. In practice, of course, for the higher velocities the consumption per hour augments in a much more rapid ratio. But the augmentation is not a fixed quantity. Every ship has a different rate. Again, we have the consumption of fuel varying with the power because some powers are more economical than others. The general result may be summed up in a few words. For every ship there is a speed at which she can steam further per ton of coal burned than she can steam at any other speed. What this speed may be can only be determined by direct experiment. It is clear, however, that, particularly in the case of warships, the higher the velocity the better. So much premised, we may proceed to the experiment.

Two of the most recent additions to the United States Navy are two *Dreadnoughts*. These are the *Delaware* and the *North Dakota*. It is claimed for them that they are the most powerful warships afloat. They are, in all respects but their machinery, sisters. Each can fire a broadside from ten 12-inch and seven rapid-fire 5-inch guns. Their displacement is

about 21,500 tons, and their nominal or contract speed 21 knots, a velocity which has been considerably exceeded by both ships. The *North Dakota* was built and engined by the Fore River Shipbuilding Company, and is propelled by Curtis turbines, developing on the high-speed trials 31,400 shaft horsepower, which drove her at 22.25 knots. The *Delaware* was built and engined by the Newport News Shipbuilding Company. She has reciprocating engines, and on her trial attained a speed of 21.563 knots with 28,578 indicated horsepower. Now, in this country the results of what are virtually competitive trials are kept secret from everyone but foreign officers. But a more liberal policy prevails in the United States, and facts are given to the world. Not only were trials made to settle speeds and powers, but also incidentally to fix the range of action of the two great ships. A keen controversy is going on between the two shipbuilding companies to settle this point. A table containing the official figures has been published, and we are told that the United States Bureau of Steam Engineering estimate that the *North Dakota* can steam about 5,000 knots, and the *Delaware* 6,000 knots, without quite emptying their bunkers—figures which do not appear to be justified by those in the table.

We now proceed to examine the facts in a little more detail; and, first, as to the consumption of fuel. Taking the main engines only, we find that at 12 knots, 19 knots and 21 knots, the *Delaware* required 15.48 pounds, 12.7 pounds and 12.9 pounds of feed water per horse per hour. The *North Dakota,* at the same speed, required per shaft horsepower 20.6 pounds, 14.25 pounds and 13.8 pounds. Even after we have made an allowance for the difference between shaft and indicated horsepower, we find that the Curtis turbine is not so economical as the piston engine, and, as was to be expected, this is particularly manifest for the lower powers. Every steam engine, whether it is a piston engine or a turbine, has a certain speed which is more economical than any other speed. In the *Delaware* the difference between best and worst is only 2.58 pounds; in the *North Dakota* it is 6.8 pounds, or nearly three

times as much. The *Delaware,* during a full-speed four-hours' trial run on the 23d of October, made 21.563 knots with 28,578 indicated horsepower and a coal consumption of 24 English tons per hour. The figures for the *North Dakota* have not reached us, but we deduce them from the published water consumption. At 21.64 knots she burned 28 tons per hour; taking the fraction of a knot higher speed into account, it may be said that the consumption — which is in American tons of 2,000 pounds—was about the same for both ships. This is assuming that the boilers evaporated 8 pounds of water per pound of coal, which is the stated evaporation. Applying the same rule to the *Delaware,* we find that the consumption was 26½ tons per hour. We may take it that at these high velocities there was no substantial difference between the ships.

Turning now to the lower speeds, we find, calculating in the same way, that at 12.24 knots the *Delaware* burned 5.26 tons per hour, and the *North Dakota,* at 12.06 knots, 5.6 tons. The builders of the *Delaware* give the consumption as 4.3 tons per hour at 12 knots. The Fore River Company do not give any figures. If the builders of the *Delaware* are right then the boilers must evaporate more than 8 pounds per pound of coal. The *Delaware* made, we are told, 21 knots with 23,600 horsepower, 19 knots with 14,560, and 12 knots with 3,520 indicated horsepower. It will be seen from the table on page 634 that at all these speeds the total weight of steam used by the *North Dakota,* including all auxiliaries, was in excess of that required by the *Delaware.* The data are, however, incomplete, in that nothing has been said—officially, at all events—about the comparative efficiency of the propellers in the two ships. The probability, nay, the certainty, is that the propellers of the *Delaware* are more efficient than those of the *North Dakota.* But it is not necessary to go far into this question. We have only a certain number of facts to go upon, and these show that so far as range of activity is concerned, the turbine is, other things being equal, inferior to the piston engine. The highest average speed attained by the *Delaware* was 21.563 knots, and that, as we have said, was got with 28,600 indicated horse-

power. The highest speed obtained with the sister ship was 21.64 knots—a scarcely appreciable difference, while the shaft power was 31,400, a difference of not less than 9 per cent. If we deduct 10 per cent. from the indicated horsepower of the *Delaware* engines we have 25,740 as the shaft horsepower, or but 82 per cent. of that of the *North Dakota*. These figures, inconclusive as they are, go to show that the propellers of the turbine ship must be much less efficient than those of the *Delaware*. This handicaps the turbine. But the turbine and the propeller are inseparable. They work together, and the naval architect must deal with them as a whole. The result of the experiment seems to be so far adverse to the turbine. But the end is not yet. The last word has not been said concerning either turbines or propellers.—"The Engineer."

SOME INVESTIGATIONS WITH A NEW FORM OF TORSION METER.

In the line of conversion of heat produced by combustion into power, there are several points at which measurements may be made to ascertain the output of the system. Those generally accepted are, in the case for instance, of a steam engine, weight of coal consumed, weight of steam generated, or of feed water delivered to the boiler, pressure of steam acting upon the pistons, combined with speed of piston, and brake horsepower delivered to the shaft. Obviously the last, where it can be measured, is the one most to be desired, for the others all suffer loss, the amount of which depends upon the efficiency of the boiler and engine, before final conversion into usefully applied power. Hence arises the importance attaching to brake horsepower. To measure this for large units is an exceedingly difficult operation, but a solution of the difficulty is found in the application of torsion meters which measure the amount of twist given to the shafting when power is being transmitted. In the case of turbine shafting, or shafting driven by electro-motors, where the torsion is very nearly

uniform, the problem has been attacked in different ways — with a greater or less degree of success. Several forms of torsion meter and the methods by which the twist of the shafting obtained from them is converted into horsepower, have been described in previous issues of "The Engineer."*

A considerable number of tests upon running shafting have been made in recent years upon ships built by Messrs. William Denny & Bros., Dumbarton, with the result that a new kind of torsion indicator has been evolved, which is simple in form, and whose management and use are within the scope of any engineer. Fig. 1 represents the Denny-Edgecombe torsion indicator now being placed on the market, as designed for shafting with uniform torsion. By its means an engineer running turbine machinery can in a few minutes ascertain the brake horsepower, or more correctly, perhaps, the shaft horsepower, being delivered by his engine. The instrument as shown was tested, along with two other types of torsion meter, upon the shafting of the Lancashire and Yorkshire and London and North-Western Railway Company's turbine steamers, *Duke of Cumberland* and *Duke of Argyll*. All the instruments were upon the same shaft during the speed trials, and the new torsion indicator gave complete satisfaction.

It consists of a split cast-iron tube, secured upon the shaft, revolving with it, and carrying at one end multiplying gear upon an arm. A corresponding standing arm is secured upon the shaft, and carries a small fine-toothed quadrant which engages with the multiplying gear on the arm of the tube. The multiplying gear has a light aluminum drum, which moves an endless flexible wire, and converts the torsional movement of the shaft, after multiplication by the small gearing, into longitudinal movement along the tube. Running easily upon the tube is a very light aluminum traveler, which is secured to the wire by a grip, and so acquires the same motion. As the torsion in the shaft is increased, the standing arm is moved relatively to the arm of the tube, which is, of course, quite unaffected by torsion, and thus the light traveler moves cor-

*"The Engineer," November 13, December 11, 1908, and January 22, 1909.

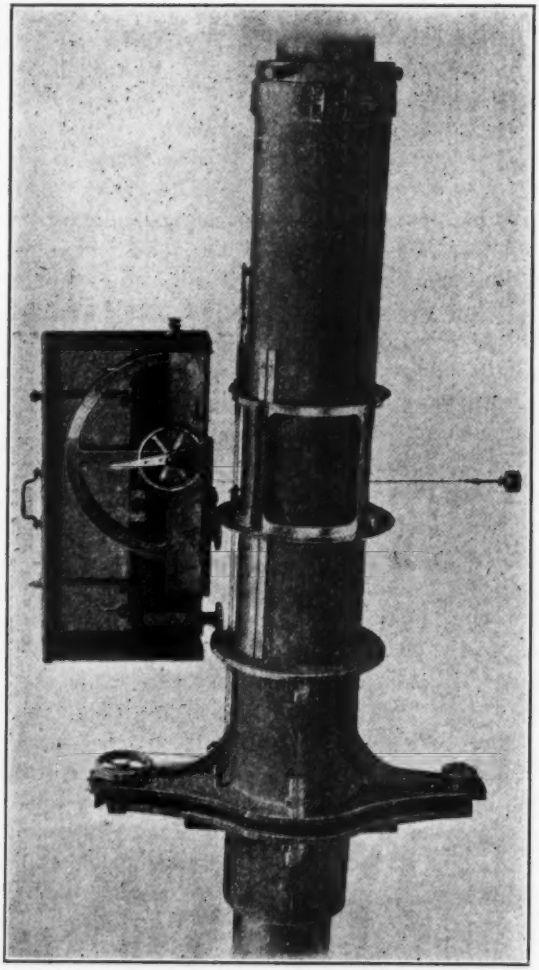


Fig. 1.—Denny-Edgecombe Torsion Meter for Uniform Torsion.

respondingly along the tube. The traveler has an accurately turned flange, which corresponds with a similar base flange cast upon the tube. It will be seen that if there be any longitudinal movement of the shafting due to expansion, or other causes,

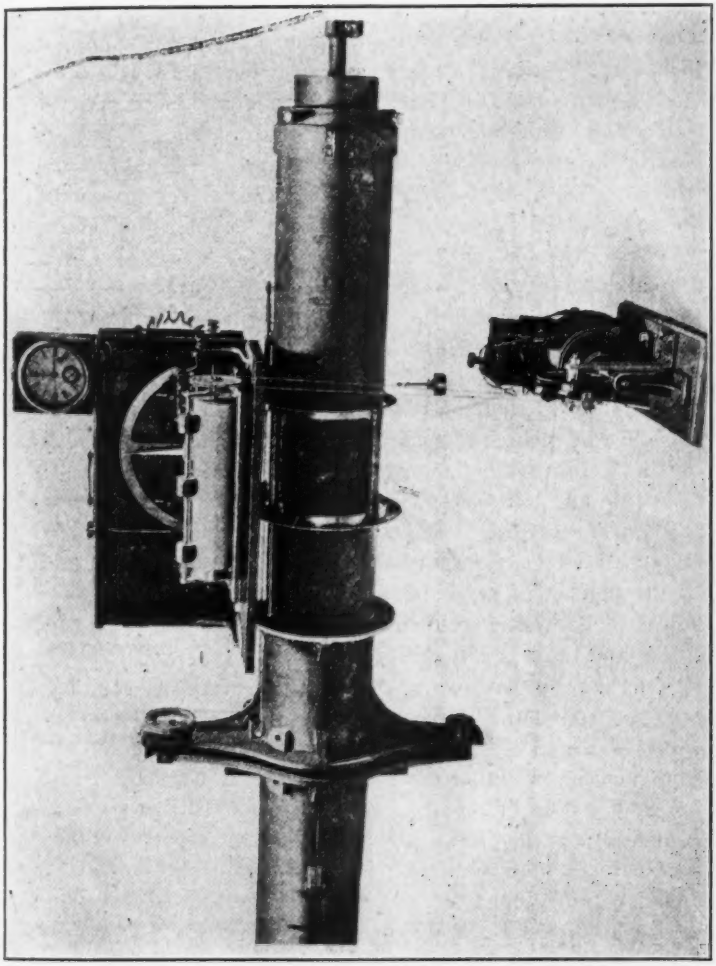


Fig. 2.—Torsion Meter Fitted with Recording Apparatus.

both the base flange and the light traveler flange move together, and the distance between them is unaltered by anything else than torsion of the shafting. The movement of the traveler flange relatively to the base flange, when multiplied by the

number of revolutions per minute, is a measure of the power. The indicator from which this movement is read is applied to the flanges by means of two small revolving wheels, which run against their turned surfaces. One of these wheels is attached to the body of the indicator, and by means of a small spring is kept in direct contact with the base flange. The other small wheel is upon a slide on the indicator, and is held against the flange by a tension weight. The slide in the indicator box, to which the wheel is attached, is connected by a flexible wire to a small drum carrying a pointer moving over a graduated scale. When the shaft is at rest, or moving very slowly without torsion, the two small wheels are held against their respective flanges, and the indicator is adjusted by means provided for the purpose, so that the pointer stands at zero. The wheels may then be secured in such a position that they are quite out of contact with the flanges, and the indicator box may be locked up and left. When it is desired to read the power being transmitted, the wheels are allowed to run each upon its respective flange. The indication read on the scale, multiplied by the number of revolutions per minute, is the shaft horsepower. The wheels are then again fixed in the out-of-gear position, and the box left locked up.

A light protective casing of wood or sheet iron should be fitted all round the instrument, except at the indicator box, which is carried on supports from the ground, or the hull of a ship, in such a way that it can be easily lifted in or out of place for examination or repair of any part, should it be necessary. A special scale has to be supplied to suit each particular case in which the indicator is applied. If the scale be divided in inches, or any other fixed unit, then the shaft-horsepower with N revolutions and a scale reading of X is $N \times X \times K$. The value of K varies for different instruments and shafts. It is most accurately obtained by a torsional test of the shaft before it is placed in position. A very close approximation to its value may, however, be made from many actual tests made with other shafts, and in any case a day-to-day comparison of the output with the steam pressure, at the high-pressure tur-

bine, for instance, can be made and logged by the engineer. The instrument is simple and easily understood; the shaft horsepower is immediately obtained either ahead or astern, and the zero position may be verified or adjusted when the shaft is stopped. Its accuracy is unaffected by lateral or longitudinal movements of the shafting in the bearings, or by any alteration in the form of hull of ships when under power; a very important matter. The wheels can be put out of gear, and only placed in contact with the revolving flanges for the few moments necessary to read the indication. Fig. 2 shows the instrument fitted with recording apparatus so that a diagram of the twist and revolutions per minute may be obtained. It is not contemplated that for every-day use this would be required, the pointer being regarded as sufficient.

This simple form of torsion indicator is the result of many experiments made with a view to discover the quality and quantity of the movements taking place in shafting when transmitting power. Obviously there is a great difference in the condition of shafting when driven by a steam turbine or electro-motor, in which case it receives many impulses per revolution; and that when driven by a reciprocating engine and receiving only two, three or four impulses per revolution. In the latter case the torsion of the shaft is always in a state of violent fluctuation, as will be seen later. It is clear, too, that the interposition of a flywheel between the engine and its work, or rapid changes of load in the work itself, produce corresponding variations in the torsion to which the shafting is subjected.

When the instrument is applied to shafting driven by a reciprocating engine it is necessary to carry the quadrant engaging the multiplying gear upon an arm composed of two laminated springs, so that the violent jarring which takes place in the shafting may not be communicated to the lighter mechanism. A different instrument was used, however, for investigating the rapid movements taking place in the shafts of marine engines driven by reciprocating machinery, a torsion meter with a countershaft being adopted in order that the traveler flange should be as small and light as possible.

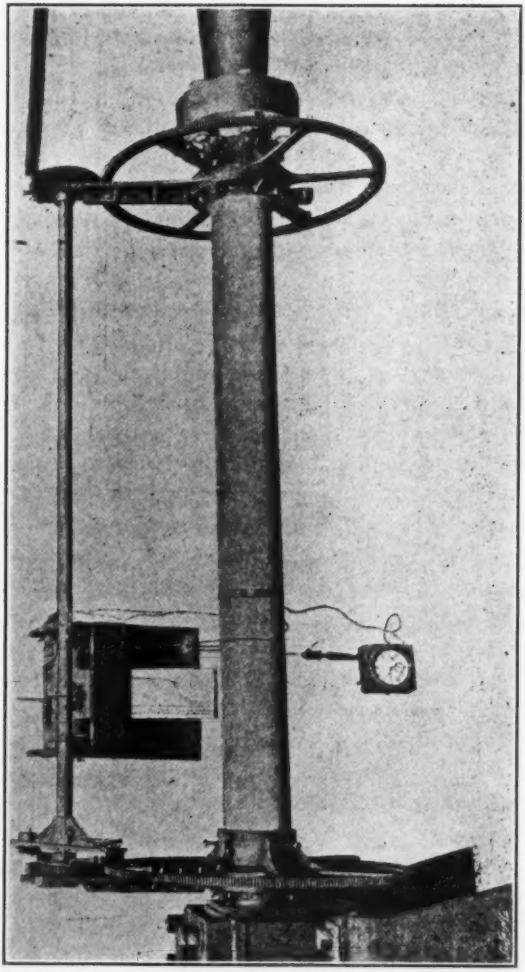


Fig. 3.—Torsion Meter for Reciprocating Engines.

Fig. 3 shows an instrument of the kind, although the toothed wheels on the shafting in the illustration were much larger than those used on the shafting of a steamer. The instrument was fitted in a vessel and tested at sea during calm

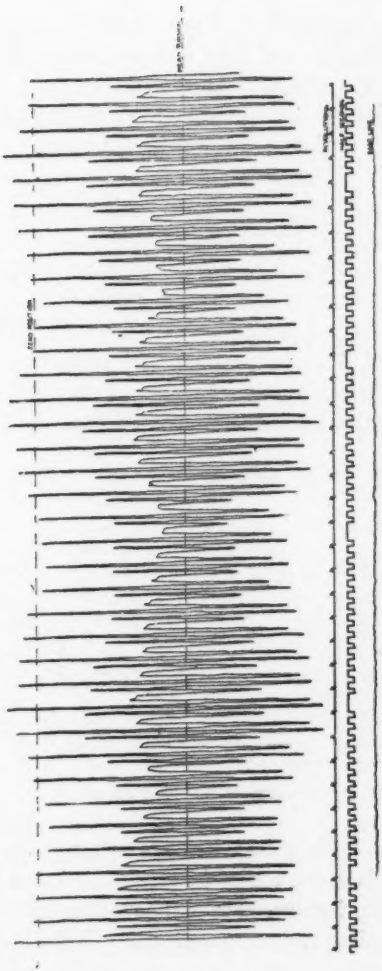


Fig. 4.—Diagram of Reciprocating Marine Engine.

weather. It has also been fitted on several ships during their trials, and some results were published in "The Engineer" in November, 1908, Vol. cvi, p. 559, showing that actual continuous diagrams of the movements had been taken with it in this country.

Fig. 4 shows the kind of diagram obtained when the ship was running steadily, and it is reproduced for the purpose of comparison with a diagram taken by the same torsion meter from the shafting of a land engine. A study of it shows that not only is there a violent reciprocating motion in the shaft all the time, but that there is a definite order and sequence in its amplitudes.

Fig. 5 shows a diagram taken rapidly during one revolution, and it is interesting to study this diagram first, and then that taken over a great number of revolutions, as in Fig. 4, when the sequence of phases produced by the synchronization of periodicity of the shaft, engine and propeller becomes evident. The propeller on the shaft from which these two diagrams were obtained was a four-bladed one. In Fig. 4 it will

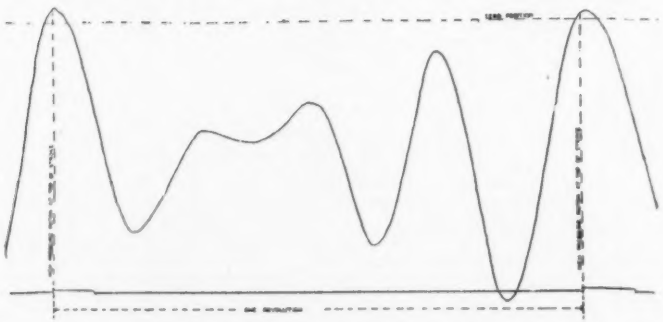


Fig. 5.—Diagram of One Revolution.

be seen that the highest impulses are attained when the high-pressure piston is at the top of its stroke, and that all the peaks of high-pressure, intermediate and low-pressure impulses can be joined in similar curves, varying in a definite relation. In Fig. 5 it is seen that something was happening of a cushioning nature at the peak of the intermediate impulse, and the engineer of the ship thought at the time that it was caused by water in the cylinder. Curiously enough, the indicator cards taken about the same time by ordinary steam indicator do not show anything abnormal—see Fig. 6.

It is evident from these diagrams that the stresses on the shaft of an ordinary reciprocating engine when running easily in calm weather are very serious. They are greatly increased in violence when starting the ship from rest, turning with the rudder hard over, or when checking the ahead motion by reversing the propellers. Even in a turbine steamer the shaft is subject to violent oscillations—caused probably by cavitation at the propeller—when reversing rapidly from full speed ahead. But this is, of course, only of occasional occurrence, and not the regular condition of things. It is not surprising that the shafting of ships with reciprocating engines has so often failed, but rather that incipient weakness is not more often developed into an actual fracture by the rapid and violent variations in the torsion which occur even under the normal

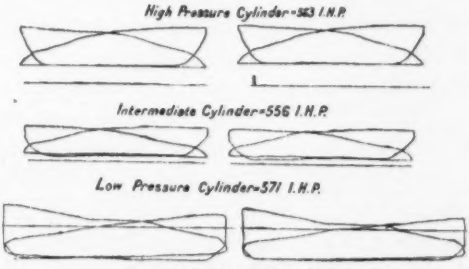


Fig. 6—INDICATOR CARDS

conditions of working. It may be safely inferred that the life of the shafting in turbine steamers will be much longer with ordinary conditions than that of the shafting of reciprocating engines.

In order further to examine the condition of mean torsion in reciprocating-engine shafting a spring arm was introduced to carry the multiplying quadrant. This had the effect of reducing the oscillations transmitted to a very small amount at the pencil. A feature of the torsion in one steamer when thus taken was that the mean torsion diagram, instead of being a straight line, was of a regular sinuous character, the amplitude of each wave covering several revolutions. The ship

was running in calm water, but there was a slight irregular vibration of the vessel, increasing and decreasing in periods which appeared to agree with the wave periods of the mean torsion. Unfortunately, this similarity of period was not noticed in time to verify an absolute agreement. It seems only

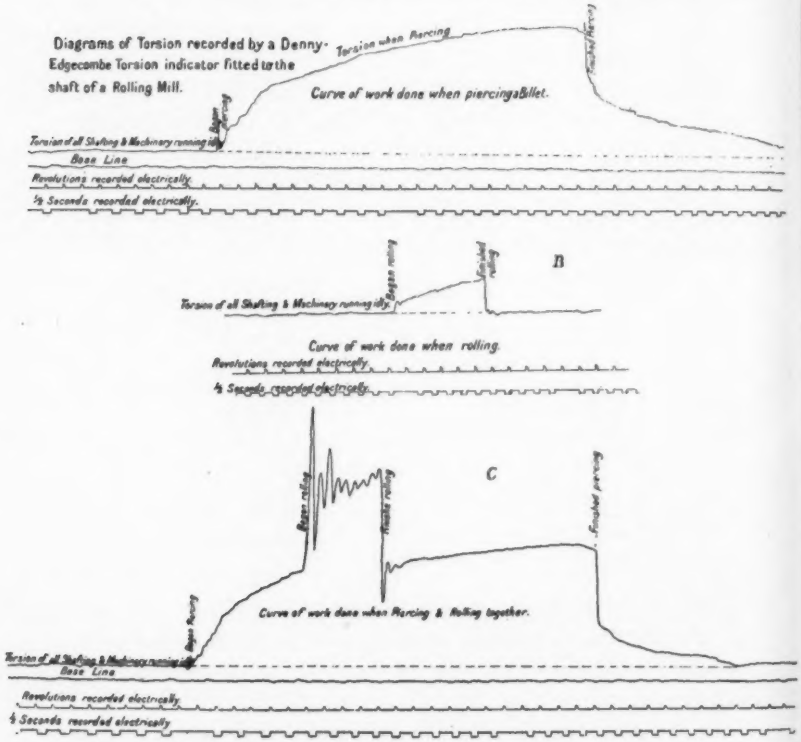

Diagrams of Torsion recorded by a Denny-Edgecombe Torsion indicator fitted to the shaft of a Rolling Mill.
Torsion when Piercing
Curve of work done when piercing a Billet.
Torsion of all Shafting & Machinery running idly.
Base Line
Revolutions recorded electrically.
½ Seconds recorded electrically.
B
Curve of work done when rolling.
C
Curve of work done when Piercing & Rolling together.

Fig. 7.—DIAGRAMS TAKEN ON A ROLLING MILL.

reasonable to assume that where vibration is set up there must be a corresponding absorption of power, and it is natural to expect that this would be shown in the torsion measurements if the power developed by the engine remained constant. The steam indicators giving a record of one revolution only, and

the four cylinders not being necessarily indicated during the same revolution, give no information which would substantiate or refute this supposition. When the new torsion indicator was being used during the trials of the turbine steamer *Duke of Argyll* at Skelmorlie, a wave was formed upon the usually straight mean torsion diagram, and it corresponded in period with a slight heaving motion of the ship felt in the tunnel at the time. This was found to be caused by the passing of a high-speed torpedo vessel also running measured-mile tests. It is likely that the torsion indicator would show similar fluctuations if used during rough weather at sea.

The instrument shown in Fig. 3 was made for investigations on a rolling-mill plant, where the conditions were of quite a different character from those of the foregoing cases. The plant tested comprised several mills all driven from one main shaft. The motive power was a horizontal steam engine, having a very heavy flywheel of large diameter, to take up the peak loads which occurred for a few seconds at some stages of the mill operations. There was a part of the main shafting between the flywheel and the first mill, about 9 feet long and 9 inches diameter, and upon this the torsion indicator was fitted, as shown in the illustration. As the operation of the mill or mills was intermittent, and only lasted for a few seconds at a time, it was a matter of some difficulty to estimate from the engine performance what the work actually done amounted to. A continuous steam-indicator diagram would have thrown some light upon the matter, but the revolutions being variable, the work done by the flywheel was more or less indeterminate. The diagrams shown in Fig. 7 were obtained from the shafting between the flywheel and the mills, and so give a complete record of the actual power transmitted. The method of working was to run the engine, flywheel and shafting up to as high a speed as desired, and when this was done, to put one or two or more of the mills into operation, when the revolutions fell rapidly as the momentum of the flywheel was absorbed. The torsion meter was arranged in such a way that the shafting drove the paper on which the diagram was taken, and by this

means an indication of the varying speed of the shaft was transferred to the diagram. The diagram thus became a record of "work" done, as well as a record of revolutions per minute, and a measure of the torsional variation. It will be seen that the character of the diagrams is quite different from that of those obtained from the shafting of a reciprocating engine with no flywheel and with a more or less regular load. The diagram was started when the speed of the engine and flywheel had been raised to its maximum and was driving all the shafting and machinery idly. As soon as the operation of piercing a billet commenced the diagram A, as recorded in Fig. 7, was obtained, the line of torsion falling back to its original position after the process ended. Diagrams are given for piercing alone, and for rolling alone, but, perhaps, the most interesting is C, in which case the two operations took place simultaneously. Here we see the "milling" load clearly added to the "piercing" load, with the peak load momentarily very much increased. Many diagrams such as these were taken, but the examples given are sufficient to show their common nature.

Where anything of an abnormal nature occurred, such as the sticking of a billet during its passage through the reducing rolls, the consequent increase of torsion was immediately recorded upon the diagram.

The application of the indicator to a rolling mill was a case in which the violent oscillations could not be damped out by a spring arm, for the reason that a complete record of the peak load was desired. The mean torsion would have given only an incomplete analysis of the operation. In spite of the rapidity and amount of the fluctuations, the instrument ran a whole day with only one failure of the light indicating mechanism, and this was repaired in a few minutes. Analyses of the records enabled a complete statement of all the particulars of time, load and power during each operation to be easily made. In the case of turbine engines fitted with the new torsion indicator shown in Fig. 1 no such difficulties as the above have to be contended with, and the process of obtaining shaft

horsepower is as simple as reading pressure from a pressure gauge. There is very little variation of torsion shown by the indicator even at the highest speeds on turbine shafting, and even if there were, there is no question of obtaining an incorrect reading, for the pointer will always indicate the mean torsion and not the torsion at any particular part of a revolution. Perhaps the most important point about it is that the whole instrument is part of the shafting and is in no way affected by any movement which is not torsion. It has appealed to those engineers who have hitherto had an opportunity of seeing it in operation as a simple and practical tool. We understand that inquiries about this torsionmeter can be obtained from Mr. F. T. Edgecombe, Comely Bank, Dumbarton, N. B. —"The Engineer."

FUEL ECONOMY AND SMOKELESS COMBUSTION.

The calorific power of fuel is determined in the laboratory by burning a weighed sample in a calorimeter. The bomb calorimeter of Messrs. Berthelot & Vieille is one of the most reliable. In this instrument, which is very simple, the powdered fuel is burned by pure oxygen under pressure in an iron vessel under water. The ignition is caused electrically, and the rise of temperature of the water gives the heating value of the fuel.

Control of Combustion.—Combustion is in reality an oxidation of the carbon and hydrocarbons of the fuel by the oxygen of the air, the result of which is heat, and it is our object when burning fuel in the furnace of a boiler to obtain this heat, conduct it through the heating surfaces of the boiler, and utilize it for the generation of steam. There are two distinct stages through which coal passes during combustion, first the distillation and combustion of the gases, and secondly the combustion of the coke or carbon that remains after the gases have been disposed of.

Take the combustion of bituminous coal, which we have difficulty in burning economically in furnaces owing to the in-

efficiency of the ordinary arrangements for consuming the gaseous products; assuming fresh coal is shoveled on a bright fire, naturally there is a sudden fall of temperature as the coal absorbs heat from the glowing and incandescent fuel, which liberates the various hydrocarbons, viz: methane or marsh gas, CH_4, ethylene or olefiant gas, C_2H_4, and acetylene, C_2H_2, and as insufficient air generally passes through the fire to ignite these gases, provision must be made for consuming them, otherwise there will be a considerable loss, as in addition to the gases passing away unconsumed, they will have taken heat from the fire to effect their distillation. If we supply sufficient air and the temperature is up to the point of ignition, we shall obtain complete combustion forming steam and carbonic-acid gas. Each volume of methane will require two volumes of oxygen

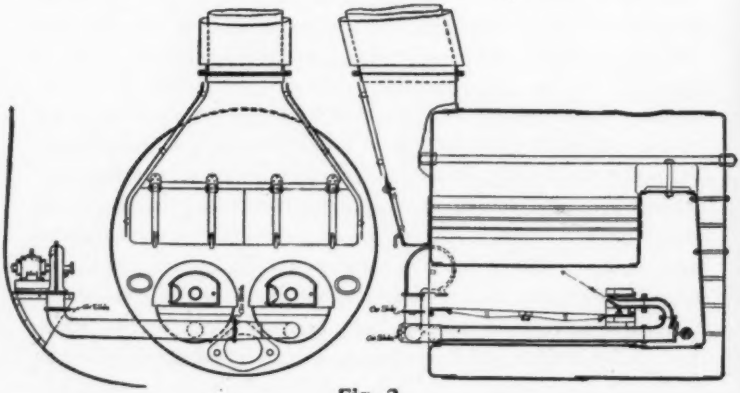


Fig. 2.

for its combustion, and each volume of the olefiant gas three volumes of oxygen, and acetylene two and a half volumes. Therefore, between two and three volumes of oxygen will be required for the complete combustion of each volume of the gas formed in the furnace; and as the oxygen in atmospheric air amounts only to one-fifth of its bulk, between ten and fifteen volumes of air will be required for each volume of gas.

Now take the coke or carbon that remains on the bars after

the gases have been disposed of: the air for its combustion should pass up between the bars and through the fuel, and the union of oxygen with the carbon is complete, and carbonic acid gas, CO_2, is formed; but if the fires are heavy and the layer of coal thick, or the bars spaced too closely together, much of this gas as it passes through the fire takes up more carbon, forming carbonic oxide gas, CO; and unless provision is made for consuming this gas by supplying additional oxygen, a very considerable amount of heat will be wasted, the products of combustion passing off as carbonic oxide, CO, by which less than one-third the heat is produced that would be yielded if the combustion were complete and the products passed off in the form of carbonic acid gas, CO_2. It is necessary, therefore, with thick fires, to admit air above the fuel for the complete combustion of the carbon in addition to the air allowed for the combustion of the gases. The average fireman is no enthusiast on the question of Fuel Economy, and his main object is to get through his watch as easily as he can; and as firing heavily and as seldom as possible entails less labor on his part, it is the course he usually adopts; great care should, therefore, be taken by those in charge to ensure that the fires are kept at a fairly thin and uniform thickness, and the bars well covered. With thick firing, although the doors are opened less frequently, large volumes of gases are suddenly produced for which there is generally not sufficient air for combustion, and an undue lowering of temperature is caused which may also prevent ignition. I have evidenced this in some makes of water-tube boilers where there is little space for the gases to mingle with the air and burn before reaching the tubes. With thin fires probably sufficient air for complete combustion passes through the fuel, in which case we are independent of the air supply above it.

It is essential for economy that the length of grate should be kept within reasonable limits to enable the fireman to keep the bars well and uniformly covered with coal, as with long grates there is danger that the back parts will be neglected and insufficiently covered, thus admitting cold air to rush in and cool the gases, tubes, etc., with a considerable loss of heat. Many boil-

ers are designed with the combustion-chamber space too restricted, thus preventing the gases from mingling and expanding; by increasing the depth of the combustion chamber and decreasing the length of furnace better results would be obtained. It should be borne in mind that no amount of heat will cause combustion of the gases unless air be supplied; but on the other hand, if the gases are not kept up to the temperature of ignition, the oxygen of the air will not chemically react with them and cause combustion to take place.

Air required for combustion.—The composition of air by weight is practically 23 parts oxygen, 77 nitrogen, hence to obtain 1 pound of oxygen $\frac{100}{23}$ pounds of air are required. From their atomic weights, it is seen that for complete combustion 1 pound of carbon would require $\frac{32}{12} \times \frac{100}{23} = 11.6$, or say 12 pounds of air, but in practice we must allow double this quantity, viz: 24 pounds, in order to insure perfect admixture. Thirteen cubic feet of air at 60 degrees F. weighs 1 pound; we shall, therefore, require nearly 700,000 cubic feet of air for the combustion of one ton of coal, about 200,000 cubic feet of which must be admitted above the bars for the combustion of the gases, as if it were allowed to pass through the burning coal on the grate it would be deprived of a great portion of its oxygen, and its value for burning the gas be depreciated. It is, however, the object of the engineer to reduce this quantity of air to a minimum, as a large amount of heat is wasted in heating it up to the temperature of the flue gases. The total amount of heat produced by the complete combustion of 1 pound of carbon is found by experiment to be 14,500 thermal units, but if the supply of air is deficient or injudiciously applied so that CO is produced, the amount of heat produced is only 4,400 B.T.U. instead of 14,500 heat units which would have been yielded if the air supply had been adequate and CO_2 formed. We see, then, the importance of admitting sufficient air to the furnace as an aid to combustion; but, on the other hand, if an excess of air is admitted the gases will be cooled and will not chemically combine and ignite; the presence of smoke, therefore, does not always indicate that there is insufficient oxygen.

Again, absence of smoke does not necessarily mean that we are burning our fuel economically, as CO like CO_2 is invisible, and we may be losing a considerable number of heat units without being aware of the fact. There are numerous coal-consuming appliances on the market designed to promote more perfect combustion, such as air-regulating devices for furnace doors, induced currents of air introduced at the bridge of the furnace, etc., and various other arrangements which have been used with more or less success. It is, however, to the bridge of the furnace that we must look as the correct place to introduce air as an aid to combustion, provided it be admitted at the right time, at sufficient temperature, and in correct proportions. The enormous volume of air that is necessary for the combustion of one ton of coal has been referred to. Much of this air in its passage through the furnace at a high velocity does not come in actual contact with the burning fuel, and gets no chance of being thoroughly mingled with the gases. These gases to a great extent pass along in contact with the comparatively cool plate of the furnace crown and do not obtain sufficient heat, and in some cases are even reduced in temperature by coming in contact with the cool plates or the colder air supplied for combustion, in consequence of which they fail to ignite before passing out of the combustion chamber, thus a considerable loss of furnace efficiency occurs.

A thin sheet of heated air admitted under pressure at the bridge directed upwards and towards the furnace door should tend to trap the gases and cause them to be thoroughly mingled and consumed, and by adopting this method we should be enabled to decrease the amount of air admitted above the bars, as it would be more thoroughly mingled with the gases. I will, therefore, bring before your notice an appliance which I have found very successful both for the prevention of smoke and for securing more perfect combustion. The apparatus is called the "Fumicide Smoke Consumer and Fuel Economizer," and has been in use on boilers which have been under my observation for the past two years with excellent results. The smoke consumer consists of a rectangular box set in the bridge of the

furnace into which a steam jet forces air, the mixed superheated steam and air being discharged through a long slit in the box back towards the furnace door. The jet is in the front of the boiler, the air and steam being carried to the box on the bridge by a cast-iron pipe. Realizing the great objection, from an economical point of view, of inducing an air blast by means of a steam jet, we have lately made experiments on one of our Cornish boilers with a fan for the purpose of creating this blast. We have erected one of the Sturtevant Engineering Company's small monogram blowing fans direct coupled to an electric motor. The accompanying sketch shows a simple method of running galvanized sheet-steel piping to the cast-iron pipe which is connected to the superheater box of apparatus. An air gate is fitted in a convenient position to enable the fireman to regulate the air supply to the smoke consumer. By adopting this arrangement we estimate roughly to have gained a saving in working cost of apparatus of 50 per cent. over that of the steam jet. We have made tests of the flue gases and obtained an average of 12½ per cent. CO_2 whilst using the fan, and 10½ per cent. CO_2 whilst using the steam jet. With the apparatus shut down altogether an average of only 6 per cent. CO_2 was obtained.—J. R. M. Fitch, in "The Steamship."

VULCAN MECHANICAL FUEL FEEDER.

This furnace stoker has but two internal-moving parts, and consists essentially of a coal hopper, with a taper bottom, in which a crushing-feeding cylinder revolves; a distributor which revolves in a chamber below the crusher, and a chain and sprocket wheel for operating the crusher and distributor. Fig. 1 shows a semi-sectional view of the feeder. It is self-contained, and is attached to the boiler front, as shown in Fig. 3.

Referring to Fig. 1 it will be seen that the fuel is fed to the first moving member or crushing-feeding cylinder by gravity from the hopper above. This cylinder is fitted with teeth, so arranged as to compensate for the adhesion of fuel to the side

walls of the machine, which insures the same rate of feed at the ends of the distributing member as takes place at the center. These teeth also furnish a positive feed, and, as a result, wet fuel can be handled just as effectually as dry fuel. The teeth are set spirally about the cylinder from end to end and are alternately long and short, as shown.

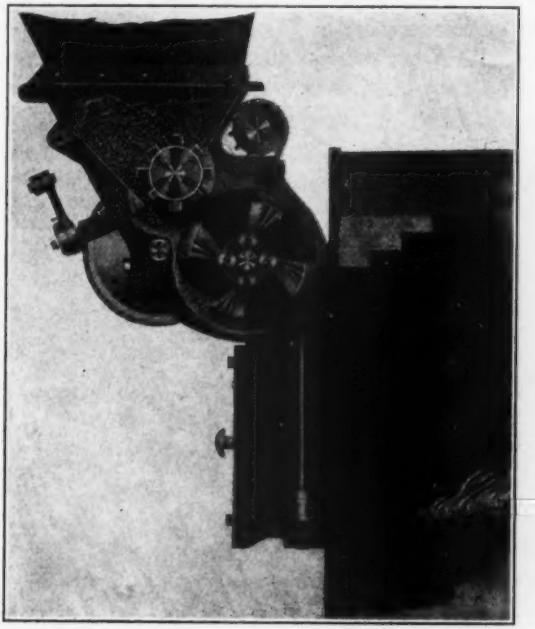


Fig. 1.—Semi-Sectional View of the Vulcan Fuel Feeder.

The second moving member, or distributor, Fig. 2, is composed of a number of blades which form a helix, each succeeding train of blades being of opposite pitch to the preceding one. This results in automatically delivering the fuel, first upon the right-hand side, and then upon the left-hand side of the grate, as it is discharged from the lip of the discharge plate. The protruding lip beyond the vertical center line and the degree of pitch of the distributing blades are varied so as to

secure proper distribution for grates of different lengths and widths. This distributor rotates as indicated in Fig. 1, at a speed of from 350 to 450 revolutions per minute.

Fuel is delivered to the distributor from the projecting lip, forming the shelf just below the crushing cylinder. The amount of fuel delivered to the distributor is regulated by the crusher jaw, which is pivoted at the upper end, as shown, and is adjusted to and from the feeding and crushing cylinder by the arm and ratchet shown on the outside of the hopper. The opening between these points on the crusher jaw and the surface of the crushing cylinder determines the amount of fuel delivered to the furnace, which is controlled from practically nothing up to a good margin above the maximum amount required.

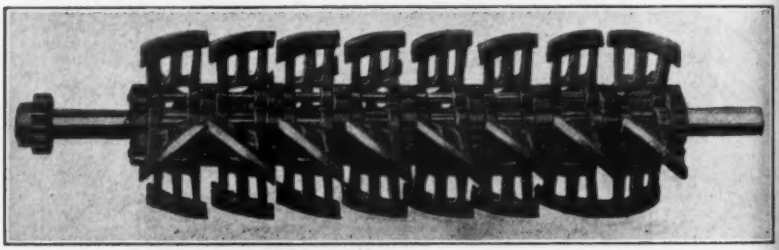


Fig. 2.—THE DISTRIBUTOR.

Grooves are provided in the crusher jaw and also in the frame casting at the rear of the crusher cylinder for the passage of the crusher teeth, when the feed of fuel is shut off, or regulated to a very fine adjustment.

When handling slack or crushed coal the crusher cylinder merely feeds the fuel to the distributing member. In case run-of-mine coal is used the lumps are engaged between the crushing teeth and the crusher jaw, where they are reduced in size and then delivered to the distributor.

The distributor blades are held in a radial position by centrifugal force, so that in case any unusual obstruction is encountered at the end of the shelf lip, the blades will swing back

on their pivot rods, which will greatly increase the clearance, when the blade following will sweep the obstruction into the furnace.

Fig. 4 shows the driving chain, sprocket wheel and a part of the interior arrangement of the crusher and distributor, also the furnace door which is fitted to the feeder frame. By means of this door the furnace is always accessible for slicing, cleaning or banking the fires, for burning of refuse or, in case of emergency, for feeding the fire by hand.

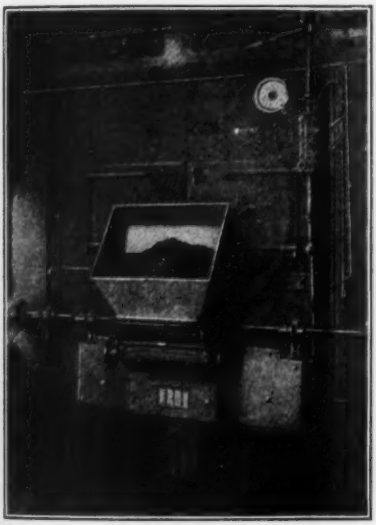


Fig. 3.—Feeder Attached to a Boiler Front.

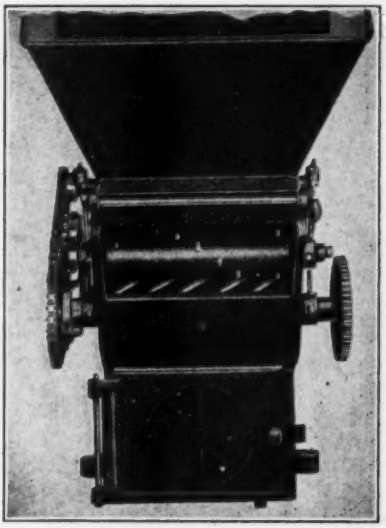


Fig. 4.—Showing Interior of Feeder and Sprocket-Gear Case.

A very favorable feature about this stoker is that it can be attached to any boiler front, with but slight alterations, the same grate can be used, the ashpit does not have to be deepened, and there is no objectionable protrusion into the boiler room.

This stoker is also available in cases where the draft is poor, because the fuel being continuously fed onto the grate allows

of a very thin fire being carried, which increases the effectiveness of the available draft. As there is no movement of the grate, loss of unburned coal sifting into the ashpit is eliminated to only that brought about by slicing and cleaning the fires. The ashpit is cleaned out in the same usual manner. When it is desired to blow the tubes of a horizontal tubular boiler, the hopper, which is hinged, can be dropped down in front of the boiler out of the way.

This fuel feeder is manufactured by the Vulcan Furnace Company, 719 Chamber of Commerce building, Buffalo, N. Y. —"Power and The Engineer."

NAVAL NOTES.

FRANCE.

The New Torpedo-School Division.—In order to assure unity of instruction among the different specialized branches pertaining to the torpedo and electrical services, a new torpedo-school division is being formed, which will be under the command of a Rear Admiral. The division will include: 1. The school for torpedo officers, which is installed on shore; 2. The school for seamen torpedo men on board the *Marceau;* 3. The school for torpedo mechanics on board the *Cécille;* 4. The central school of instruction for the heads of the wireless-telegraphy stations at Toulon; 5. The school ships on board which the torpedo apprentices are being trained. The Central School of Instruction for Wireless Telegraphy at Brest was suppressed on the 1st November. Rear Admiral Guillou, who has been appointed to the command of the new Torpedo Division, has hoisted his flag on board the battleship *Brennus,* while Rear Admiral Le Bris, who has been appointed to the command of the Gunnery School Division, has his flag flying in the battleship *Masséna;* both these ships are to be kept fully manned and ready for service.

Launch of the "Mirabeau."—The new first-class battleship *Mirabeau* was successfully launched at Lorient on the 28th of October last. She will be the fifth of the six battleships of the 1906 program to take the water; of the four already launched the *Voltaire* was built at La Seyne, the *Diderot* and *Condorcet* at Saint-Nazaire, the *Danton* at Brest, and the sixth, the *Vergniaud,* is being built at Bordeaux. The five that will have been launched in 1909 represent a total of 91,590 tons, the most important and largest amount ever launched in France in one year; it is nevertheless inferior to the tonnage launched in

Germany in 1908-9, during which period 104,971 tons of new ships were constructed.

Recommendations of the Conseil Supérieur de la Marine.—The French fleet, according to the Superior Naval Council, should consist of 45 battleships, 12 scouts, 140 torpedo-vessels, 64 submarines and 3 mine layers.

No more armored cruisers should be biult; these vessels no longer differing from battleships, except in having fewer heavy guns and more speed, the Council has decided that they are unnecessary. Such armored cruisers as already exist should be utilized in divisions on foreign service.

The scouts should be lightly armed vessels, but of sufficient size to keep the sea in any weather. and of great speed—5,000 to 6,000 tons at least.

There should be two classes of torpedo vessels; one class, for use with the fleet, should be of at least 600 tons displacement; the other, constituting the coast flotillas, should not exceed 300 tons displacement. The Council proposes 60 of the first class and 80 of the second.

The submarines should all be of great radius of action and designed for offensive purposes, the inshore defence of harbors being assured by the older offensive boats and no longer by boats specially constructed for defensive purposes.

The Council, furthermore, has called attention to the necessity of constructing a special-practice ship for midshipmen, and of providing appropriate material for raising and dragging for submarine mines; the recent purchases of steam trawlers by the British Admiralty may serve as a guide in this respect.

As regards the fighting ships of all classes, the Council expressed the wish that the proposed program be completed in 1925 at the latest. It has fixed as age limits for battleships 20 years and for scouts and torpedo vessels 15 years (there is insufficient experience as yet to fix an age limit for submarines).

According to these figures it will be necessary to lay down, from 1910 to 1922, inclusive, 39 battleships, or 3 a year; 12 scouts, or 1 a year; 55 squadron torpedo vessels, or 4 a year;

75 flotilla torpedo vessels, or 6 a year; and about 40 submarines, or 3 a year. Taking all these vessels at the average displacements now adopted, we arrive at an annual expenditure of 230,000,000 francs (£9,200,000) for new constructions, instead of the 120,000,000 (£4,800,000) allowed in recent years.

Regarding naval stations the Council expressed the opinion that that of Rochefort should be entirely given up; that Lorient should become merely a building yard for large ships; that Cherbourg, Toulon and Brest should remain as they now are, yards for building and repair, the former building only submarines and torpedo vessels; the last only large ships. It recommended that Bizerta be fitted as a principal base of operations, on the same footing as Brest, Cherbourg and Toulon, and that each of these ports have at least four docks capable of taking the largest ships of the fleet. It insisted that at least three-fourths of all vessels be constantly in commission with full complements, and that the number of men under enlistment be always amply sufficient for all necessary services in addition to the manning of these vessels.

Finally, the Council, withdrawing from the position which it took in 1907, declares itself in favor of unity of caliber and gives the preference to a main armament composed solely of 12-inch guns.

The characteristics decided upon by the Council for the battleships, which are to be asked for from the legislature, with authority to begin them in 1910, are now known.

The main armament consists of twelve 12-inch guns, distributed in 5 turrets; two 3-gun turrets on the middle line, one forward and the other aft, and three 2-gun turrets, of which one is on the middle line, just forward of the after turret, and the two others on the broadsides. The after double turret does not fire over the triple turret. In short, except for the introduction of the 3-gun turrets, the general arrangement is that of the *Dreadnought*. Ten guns can be fired on either broadside and 7 ahead or astern.

For three or four years past 3-gun turrets have been talked of for projected battleships; there has been talk of adopting

them in Germany, and in England, but hitherto the plan has not been put into effect. Its evident advantage is allowing the placement of the same number of guns upon a less length of ship and with less weight. The no less evident disadvantage is is that in the case of any damage done the turret or its mechanisms 3 guns are put out of action instead of 2; but that is a reason which no longer is considered important; the fear of putting too many eggs in one basket has gone out of fashion. and people are no longer frightened by big guns, nor by big turrets, nor by big displacements.

Still there are many difficulties in the working out of a 3-gun turret which, although not insurmountable, are none the less serious. The right and left guns when fired will disturb the train much more than is done by the guns of double turrets, since they are further out from the axis; but it is expected that this effect will be lessened by the greater inertia of the heavier turret. The loading and other mechanisms of each gun must be completely independent, from the magazine to the line of sight, so as to allow each piece to fire the two shots a minute that can be obtained from it on the proving ground, that is, for the turret, one shot in at least every 10 seconds; and it is not very easy to assure correct pointing in so short an interval; nevertheless, thanks to the use of apparatus for continuous pointing—inspired by those which are in use in England—it is expected to succeed in attaining these conditions. Besides this main battery the Council has recommended a secondary battery of eighteen $5\frac{1}{2}$-inch guns, distributed in 3 groups, in armored casemates. Perhaps some 65-mm. guns will also be added for use against torpedo boats, although the $5\frac{1}{2}$-inch guns are intended for that purpose.

There will also be four underwater torpedo tubes of 45-cm. ($17\frac{3}{4}$-inch), two forward and two aft. The protection will be the same as in the *Danton* class, except that the funnels and ventilators will be covered with 18-cm. ($7_{1}^{1}{}_{2}$-inch) plating, as will also the casemates of the $5\frac{1}{2}$-inch guns. Large-capacity explosive shells with no perforating power are not to be allowed to destroy the superstructures and cause a loss of speed.

The famous anti-torpedo armor, which casts 1,000 tons of displacement in the *Danton* class, and which was adopted for them without preliminary test, is abandoned, although the question of its efficiency is no more settled than it was before. The test made at Lorient on a caisson representing the hull of the *Mirabeau* does not seem to have given very conclusive results, and, moreover, the system in principle was abandoned without waiting for that test. It is a decision which might advantageously have been taken three years earlier.

On the other hand, copying foreigners, we are to go back to the anti-torpedo nets. It is to be hoped that an arrangement of them will be found which will avoid the criticisms formerly made against them and which led to their abandonment fifteen years ago.

The propelling machinery will consist of four turbine ma chines like those of the *Danton*, with four screws. The maximum speed will be 20 knots.

All these characteristics lead to a displacement of 22,000 tons, like that of the German *Oldenburg* class, which have nearly the same armament (twelve 30-cm. (12-inch) and fourteen 5.9-inch), superior to that of the English *Neptune*, which is only of 20,250 tons, carries ten 12-inch guns, and steams 22 knots.

In comparison with the *Danton* class, and above all, with the hybrid plan which the Council proposed in 1907, there is evidently a sensible gain in offensive power. But there is also a 4,000-ton increase of displacement and an increase of price, which may be taken at 12,000,000 francs (£480,000) per ship. And this increased expense will probably not be without influence on the reception which will be given to the Council's projects by the Minister of Finance first and by Parliament afterwards.—"Le Yacht" ("Army and Navy Journal").

French Battleships.—The *Diderot* and *Condorcet* are two of the six battleships of the *Danton* class authorized in the French naval program of 1906-1907. The other ships of this class are the *Voltaire, Vergniaud* and *Mirabeau*. These ships are all of the same general dimensions and arrangement but differ some-

what in details. The first of these to be launched was the *Voltaire,* in January, 1909, at La Seyne, near Toulon. The *Diderot* and *Condorcet* were launched at St. Nazaire, the former on April 19, 1909, from the yards of the Chantiers de L'Atlantique, and the latter, April 20, from the Ateliers & Chantiers de La Loire. The other three ships are still on the ways, the *Verginaud* at the Ateliers de La Gironde, Bordeaux; the *Danton* at Brest, and the *Mirabeau* at Lorient. All, however, will soon be ready for launching.

The principal dimensions of the *Diderot* and *Condorcet* are as follows: Length over all, 481 feet; length on waterline, 476 feet; extreme beam, 84 feet 8 inches; depth at full load, amidships, 27 feet 1 inch; full load displacement. 18,235 tons; designed horsepower, 22,500; speed, 19.25 knots.

The ships will be propelled by four screws, each driven by Parsons turbine engines. There are eight turbines in each ship, four for full speed forward, two for the lower cruising speeds, and two for astern speed. The dimensions of the high-speed turbines are as follows: High-pressure, diameter, 9 feet; length, 23 feet 10 inches; low-pressure, diameter, 12 feet; length, 23 feet 6 inches. The high-pressure turbines drive the outside propellers, and the low-pressure turbines the inside screws. At full speed, the designed number of revolutions is 300 per minute. The high-pressure cruising turbines are on the inside shafts, the steam being exhausted into the high-pressure high-speed turbines and then into the low-pressure turbines. The cruising turbines are 8 feet 4 inches in diameter and 17 feet 6 inches long. The astern turbines drive the outside propellers and are 9 feet in diameter and 12 feet long. The inner shafts are located 7 feet from the center line of the ship, and the outer ones 21 feet 10 inches from the center line.

The *Diderot* and *Condorcet* will each be fitted with twenty-six Niclausse water-tube boilers, each boiler having 1,560 square feet of grate surface and 14,150 square feet of heating surface. The normal steam pressure will be 257 pounds per square inch. Their total cost each estimated at $10,300,000.—"International Marine Engineering."

The Superior Council of the Navy.—Under the new organization, the Council is at present composed of the following officers: Inspectors General, Vice Admirals Caillard, Germinet, Marquis and Philibert; as Chief of the Staff, Vice Admiral Marin-Darbel and Vice Admiral Jauréguiberry, the only flag officer who has left command of a squadron within the previous two years.

Reorganization of the Technical and Other Committees.—A decree in the Journal Officiel also modifies the composition of the Technical Committee of the Navy, and re-establishes the Permanent Committee for trial of ships and for control and commissioning of ships.

The Technical Committee is to consist of

1 Vice Admiral (President).
2 Rear Admirals.
3 Superior officers of the Navy.
1 Superior officer mechanician.
1 Superior officer of Naval Artillery.
2 Superior officers, Naval Engineers.
1 Lieutenant as Secretary.

Commission Permanent d'essais will consist of

1 Rear Admiral (President).
1 Superior officer of the Navy.
1 Superior officer, Naval Engineer.
1 Superior officer of Naval Artillery.
1 Superior officer mechanician.
1 Lieutenant as Secretary.

Commission Permanent du Réglement d'armement:

1 Rear Admiral (President).
2 Superior officers of the Navy.
1 Superior officer, Naval Engineer.
1 Principal mechanician.
1 Superior officer of Naval Artillery.
1 Principal Commissary.
1 Superior officer, Naval Medical Service.

Creation of a Naval Artillery Corps.—The Chamber, without discussion, has passed the *project de loi* introduced by the Minister of Marine for the creation of a Naval Artillery Corps, *corps d'ingénieurs d'artillerie de la marine.* The new Corps, which takes the place of the old Corps, officered from the Colonial Artillery, will consist of 3 general officers, 52 superior officers, 40 engineers of the rank of captain, and an indefinite number of engineers of the rank of lieutenant as may be found necessary. The engineers of the Naval Artillery Corps will be recruited from lieutenants in the Navy, naval engineers, the land artillery, and the Colonial artillery, and the officers of the new extinct Corps of Naval Artillerists, who will form the nucleus of the new Corps.

Report of Rear Admiral Le Pord.—In his report of proceedings, Rear Admiral Le Pord, who commanded the 2d Division of the 1st Squadron sent to New York for the Hudson-Fulton celebrations, after praising the hospitable and cordial way in which officers and men were received by the American Navy and the people of New York, goes on to say: "Our three battleships, *Justice, Liberté, Vérité,* nicknamed by the Americans the three bulldogs, produced the best impression, and in fact by its power and homogenity was the most imposing group of all the foreign divisions." He adds that in contrast with the embroidered full-dress uniforms of other nations, the French frock coat, even with epaulettes, had a poor appearance, and does not seem a suitable dress for ceremonial occasions such as those in which the officers had to take part.

New Docking Regulations.—The Minister of Marine has directed that different classes of ships are in future to be docked at the following periods, and that in all cases the time in dock is to be as short as possible.

1. Torpedo boats, destroyers and submarines, 7 to 9 months.
2. Cruisers and avisos, 10 to 12 months.
3. Battleships, armored cruisers and transports, 15 to 18 months.
4. Wooden and coppered vessels, 24 to 26 months.

New Trial System of Work in the Dockyards.—The Under-Secretary of State for the Navy has ordered the trial of a new system of work at Lorient Dockyard, which is an interesting experiment. It is of the nature of sub-contracts by the men themselves. Groups of workmen will be allowed to tender for certain work at an agreed price, the sole condition being that the hours of labor are not to exceed the regulation 8 hours of the workman's day. If the system works satisfactorily and gives an increased ratio of production, as is hoped, it will be extended to other dockyards and naval establishments.

New Distribution of Submarines.—The Minister of Marine has made a new distribution of the submarines, and their stations in future will be as follows:

Calais.—One group (offensive): *Germinal, Ventôse, Pluviôse.*

Cherbourg.—Two groups (offensive): *Floréal, Prarial, Messidor, Rubis, Emeraude, Opale* (shortly to be replaced by the *Archiméde*). One group (defensive): *Phogne, Naïade, Ludion, Méduse.* Spare submarine: *Francais.*

Brest.—One group (offensive): *Watt, Cugnot, Fresnel.* One group (defensive): *Triton, Silure, Espadon, Siréne.* Spare submarine: *Aigrette.*

Toulon.—One group (offensive): *Monge, Topaz, Saphir, Cigogne.* One group defensive, namely, Five submarines of 70 tons.

Bizerta.—One group (offensive): *Papin, Circé, Calypso.* One group (defensive): *Korrigan, Gnome, Follet,* plus two submarines of 700 tons.

Oran.—One group (offensive): Three submarines of the *Fructidor* class.

The remaining submarines will be placed in the category "on trial" or in "reserve."

This new disposition creates two new groups of submarines, 1st at Brest, at the entrance to the Channel, 2d at Oran, in the narrow part of the Mediterranean Sea.—"Le Temps" and "La Vie Maritime."

GERMANY.

Steam Trials.—The new first-class battleship *Nassau* and the new first-class armored cruiser *Blücher* were commisioned on the 1st of October at Wilhelmshaven for their trials, which have since been proceeding successfully. Full details as to the results of the trials are not yet forthcoming, but it is reported that at the full-speed trials the *Nassau* maintained a mean speed of 20.5 knots, being a knot and a half over the contract, and that the *Blücher* attained a speed of 25.33 knots, or more than 3 knots over the contract speed.

Three new second-class cruisers have been undergoing their trials with marked success. The *Emden,* on her full-power run, made an average of 25 knots; this was the contract speed. She has reciprocating engines, but is the last of her class which will be so fitted, as turbine engines are to be fitted in all her successors. Her sister ship, the *Dresden,* which has Parsons turbines, averaged 27.5 knots on her full-speed trials, 28 knots being reached during one run in deep water; this is 3 knots over the contract speed. Another ship of the class, the *Mainz,* which is still on her trails, has also recently averaged 27 knots on her full-speed run. All these vessels have a displacement of 3,600 tons, the contract speed being for 25 knots.

Cost of the Fleet.—The cost of the Fleet in commission is distributed as follows:

1.—*The High-Sea Fleet.*	1909. Marks.	£	s.	1908. £	s.
Active formation.....	11,779,130 =	(588,956	10)	550,135	2
16 Battleships.					
Reserve formation.....	1,146,540	(57,327	0)	61,029	0
1 Battleship.					
2 Coast-defence armored ships.					
Scouting ships.....	6,062,810	(303,140	10)	281,878	14
4 Large cruisers.					
6 Small cruisers.					
Tender.					
Torpedo vessels.....	4,764,195	(238,209	15)	189,138	16
Total.....	23,752,675	(1,187,633	15)	1,082,181	12

Showing an increase of 2,109,043 marks (£105,452 3s.) as compared with last year.

2.—*Ships on Foreign Service.*

	1909.			1908.	
	Marks.	£	s.	£	s.
The East Asian station.....	3,862,878	(193,143	18)	202,516	8
1 Large cruiser.					
3 Small cruisers.					
4 Gunboats.					
3 River gunboats.					
1 Special-service steamer.					
The Australian station.....	337,320	(16,866	0)	12,060	0
1 Small cruiser.					
The West African station.....	409,740	(20,487	0)	22,285	16
1 Small cruiser.					
1 Gunboat.					
The East African station.....	534,120	(26,706	0)	24.492	0
2 Small cruisers.					
The American station.....	504,840	(25,242	0)	23,902	16
1 Small cruiser.					
Constantinople.....	92,500	(4,625	0)	4,681	4
1 Stationaire.					
Surveying duties.....	161,880	(8,094	0)	7,880	8
1 Surveying ship.					
Total.....	5,903,278	(295,163	18)	297,818	12

Showing a decrease of 53,094 marks (£2,654 14s.) as compared with last year.

3.—*Training Ships.*

Cadets' and boys' training ships....	3,091,800	(154,590	0)	131,080	18
Gunnery-school ships.....	1,807,717	(90,385	17)	80,700	5
Torpedo-school ships.....	1,459,650	(72,982	10)	60,712	2
Submarine mining.....	661,060	(33,053	0)	30,816	0
Coast pilotage.....	61,740	(3,087	0)	2,991	2
Total.....	7,081,967	(354,098	7)	306,300	7

Showing an increase of 955,960 marks (£47,798) as compared with last year.

4.—*Special Service Ships.*

Imperial Yacht *Hohenzollern*.....	529,620	(26,481	0)	20,184	0
Surveying ships (home waters).....	70,080	(3,504	0)	2,212	16
Artillery experiments.....	653,136	(32,656	16)	33,135	0
Torpedo experimental ship	937,200	(46,860	0)	37,651	4
General experiments.....	229,170	(11,458	10)	10,936	4
Fishery protection.....	147,750	(7,387	10)	8,429	3
Total.....	2,566,956	(128,347	16)	112,548	7

Showing an increase of 315,989 marks (£15,799 9s.).

	1909. Marks.	£	s.	1908. £	s.
5.—*For Special Purposes*,	992,115	(49,605	15)	37,172	0

Showing an increase of 248,675 marks (£12,433 15s.).

COMBINED TOTAL.

High-sea fleet	23,752,675	(1,187,633	15)	1,082,181	12
Foreign service	5,903,278	(295,163	18)	297,818	12
Training ships	7,081,967	(354,098	7)	306,300	7
Special-service ships	2,566,956	(128,347	16)	112,548	7
Special purposes	992,115	(49,605	15)	37,172	0
Total	40,296,991	(2,014,849	11)	1,836,020	18

Showing an increase of 3,576,573 marks (£178,828 13s.) as compared with last year.

—"*Etat für die Verwaltung der Kaiserlichen Marine*," 1909.

Launches.—On the 30th of September last the new first-class battleship *Ostfriesland* was successfully launched from the Imperial Dockyard, Wilhelmshaven. She is the second of the improved *Dreadnought* type of battleship to take the water, the first of the type, the *Helgoland*, having been launched a few days earlier (the 25th September) from the Howaldt Yard at Kiel. The *Thuringen*, the third of the class, was launched on the 29th November from the Weser Yard, Bremen, so that Germany has now seven battleships of this formidable type afloat, and an eighth, the Ersatz *Frithjof*, building by the Schichau firm, at Elbing, is to be launched before the end of the year.

Two of the first group of four of this formidable class of ship, the *Nassau* and *Westphalen*, have already been taken over by the authorities, and as soon as their trials are completed, will be attached to the First Squadron of the High-Sea Fleet; the two others of this group, the *Posen* and *Rheinland*, are to be ready for service during the coming spring, but the *Helgoland* and her two sisters will not, it is expected, be ready before the summer of 1911. These eight battleships—including the *Ersatz Frithjof*—will form the new First Squadron of the High-Sea Fleet, and are to be stationed at Wilhelmshaven. All these eight ships are fitted with reciprocating engines, so that for steaming and evolutionary they will form a practically homogeneous squadron.

Although official details are not as yet forthcoming, yet it is believed that the dimensions of the *Helgoland* and her three sisters are as follows: Length, 456 feet; beam, 88 feet 6 inches; draught, 27 feet, with a displacement of 22,500 tons. The armament to consist of twelve 11-inch—12-inch are mentioned in some quarters—twelve 5.9-inch and twenty 3.4-inch guns, with five submerged torpedo tubes. The reciprocating engines are to develop 22,500 I.H.P., giving a speed of 19 knots, which, as the trials of the *Nassau* are showing, is likely to be appreciably exceeded. The arrangement of the main battery guns is still unknown; but, according to "La Vie Maritime," the guns will be mounted in six 2-gun turrets, four on the center line (with Nos. 2 and 3 higher than Nos. 1 and 4, so as to fire over them) and two on the broadside (one on each side) *en échelon*.

The following vessels, exclusive of destroyers, are at present under construction:

Name.	Displacement.	Where building.	Remarks.
Battleships:	*Tons.*		
Nassau	18,500	Wilhelmshaven.	Under trials.
Westphalen	18,500	Weser Yard, Bremen.	Under trials.
Rheinland	18,500	Vulcan Yard, Stettin.	Launched Sept. 24, '08.
Posen	18,500	Germania Yard, Kiel.	Launched Dec. 12, '08.
Helgoland	22,500	Howaldt Yard, Kiel.	Launched Sept. 25, '09.
Ostfriesland	22,500	Wilhelmshaven.	Launched Sept. 30, '09.
Thuringen	22,500	Weser Yard, Bremen.	Launched Nov. 29, '09.
Ersatz Frithjof	22,500	Schichau Yd., Elbing.	Building.
Ersatz Hildebrand	22,500	Imperial Yard, Kiel.	Building.
Ersatz Heimdall	22,500	Vulcan Yard, Stettin.	Building.
Armored Cruisers:			
Blücher	15,000	Imperial Yard, Kiel.	Under trials.
Von Der Tann	19,000	Voss and Blohm, Hamburg.	Launched Mar. 20, '09.
G	22,000	do.	Building.
H	22,000	do.	Building.
Protected Cruisers:			
Mainz	4,300	Vulcan Yard, Stettin.	Under trials.
Köln	4,500	Germania Yard, Kiel.	Launched June 7, '09.
Augsburg	4,500	do.	Launched July 10, '09.
Ersatz Falke	....	Where building uncertain.	.....
Ersatz Bussard	....	do.	.....

It is reported that the new battleships of this year's program are to be fitted with turbine engines. Two of these ships, the *Ersatz Hildebrand* and the *Ersatz Heimdall* have already been laid down; the *Ersatz Hildebrand* being under construction in the Imperial Dockyard, Kiel, and the *Ersatz Heimdall* at the Vulcan Yard, Stettin.

—"Marine Rundschau" and "La Vie Maritime."

ITALY.

The Annual Prize Firing.—This year's prize firing took place, as usual, in Aranci Bay, between the 23d July and 12th August. The umpires were Rear Admiral Avallone, as chief. with two captains, two commanders, and six captain-lieutenants as his assistants. The practice consisted of individual firing by the gun layers, director firing by individual ships and collective firing by divisions of three ships. At the gun-layers' practice, from two to four rounds, with practice ammunition, were fired per gun from the heavy and medium-caliber guns at a range of from 2,400 to 4,000 yards against an anchored 21 feet by 55 feet target, and from the light guns six rounds per gun with practice ammunition at a range of from 1,250 to 1,600 yards, against an anchored 9 feet by 30 target. The speed for passing the target for the large ships was fixed at 14 knots, for the torpedo craft 20 knots. The only shots counted were those that struck the target. The director firing was carried out in two series, in the first with all guns down to the 4.7-inch inclusive, against a 21-foot by 75 canvas target, erected on the hull of a torpedo boat, which was towed at a speed of 20 knots at a range of 7,000 yards; in the second, the firing was from the medium-caliber guns, the target being towed at a speed of 14 knots. The divisional firing was carried out under similar conditions, the ships being either in single line ahead or quarter column, two cables apart from each other.

The Points of Merit were as follows:

Names of ships taking part in the competition.	Gunlayer's test.		Director firing.	
	Heavy and medium caliber guns.	Light guns.	Single ships.	Divisional.
Battleships and Cruisers.				
Regina Margherita (flagship of Commander-in-Chief)	63	43	6	33
Benedetto Brin	57	59	33	
Regina Elena (flagship of second in command)	54	27	0	19
Vittorio Emanuele	50	29	35*	
Napoli	83	63	15	
Garibaldi (flagship of cruiser division)	90	89	25	24
Varese	61	71	0	
Francesco Ferruccio	91	45	28	
Torpedo Vessels.				
Agordat	...	37		
Coatit	...	32		
Espero	...	47		
Borea	...	41		
Aquilone	...	26		
Lanciere	...	55		
Granatiere	...	25		

*The *Vittorio Emanuele* fired two rounds, both hits, from her two 8-in. turret guns in sixteen seconds.

The director firing compares badly this year with last, a fact due in a great measure to the more difficult conditions imposed this year, and also to the fact that all the gunnery officers of the ships had been newly appointed.

Speed Trials.—The new first-class battleship *Napoli,* at her three-hours' full-speed trial, made an average of 22.5 knots, with the engines developing 19,700 I.H.P., and making 122 revolutions; she has a displacement of 12,500 tons, and steam is provided by twenty-two Babcock & Wilcox water-tube boilers. The new first-class battleship *Roma* has also completed her trials successfully; at her three-hours' full-speed trial she averaged 22 knots, with her engines developing 21,000 I.H.P., steam being supplied by eighteen Babcock & Wilcox water-tube boilers.

The new first-class armored cruiser *Pisa* has also successfully completed her trials; at her three-hours' forced-draft trial on the 3d July, she averaged a speed of 23.5 knots, with

the engines developing 20,812 I.H.P. and a coal consumption of 819 gr. (1 pound 10 ounces) per indicated horsepower per hour, and during a six-hours' run she maintained a speed of 23.85 knots, or nearly 2 knots over the contract. She is provided with twenty-two Belleville water-tube boilers.

A sister ship of the *Pisa*, the *Amalfi*, has also completed her trials; at her three-hours' full-speed run with the engines developing 20,812 I.H.P. and a coal consumption of 819 gr. (1 pound 10 ounces) per indicated horsepower per hour she maintained a speed of 23.64 knots. At her economical coal-consumption trial with eight boilers alight, the engines developing 2,447 I.H.P., and making 65 revolutions, she maintained a speed of 11 knots with a coal consumption of 806 gr. (1 pound 9½ ounces) per indicated horsepower per hour, and a total consumption of 1,972 kg. (2 tons 400 pounds); she has thus a steaming radius of 11,000 miles at 10 knots speed, as her total coal stowage is 2,100 tons. Steam is supplied by twenty-two Belleville boilers.—"Marine Rundschau, Rivista Marittima," and "Revue Maritime."

Torpedo Experiments against the "Francesco Morosini."—To test the actual result of torpedo fire against a battleship the Italian Navy has been making experiments with the 11,000-ton battleship *Francesco Morosini*, which has been lately struck out of the active list. Advantage was also to be taken of the experiment to test some new theories as to the best method of construction of a vessel's underwater framing and arrangement of compartments in order to best resist torpedo attack. The attack was to be made by a standard torpedo with a normal charge, and the explosion was to be effected, as far as possible, under the usual conditions of an ordinary hit. A frame cradle was attached to the vessel to carry the torpedo, which was placed at right angles to the line of keel, and with its nose touching the vessel. The regulation type and quantity of explosive were adopted, and the reservoir chamber of the torpedo was filled with compressed air of the pressure that would theoretically remain after a run of normal length. The depth below the water line was three meters. The effect of the explo-

sion above water was not more than had been expected, but it was very evident that considerable damage had been done to the vessel's hull, as she quietly took a list and, swerving over on her side, went to the bottom, where she still remains, while every day she settles still more. An examination of the damage has been made as far as possible by divers, and it appears that there is a hole of over fifty square meters in area, which is considerably more than had been expected. It is considered that the result of this experiment will be the building by Italy of a new type of vessel, very swift and with a main armament consisting of torpedoes, torpedo tubes being fitted in a very much larger quantity than in any existing vessel.—"Army and Navy Journal."

The Italian Battleship Roma.—Two first-class battleships have recently been added to the Italian Navy, both built on the same general lines, but differing in certain particulars. They are the *Napoli,* built in the navy yard at Castellamare, di Stabia, and the *Roma,* built in the navy yard at Spezia. The differences are more or less minor in character, and the description will be confined to the *Roma.*

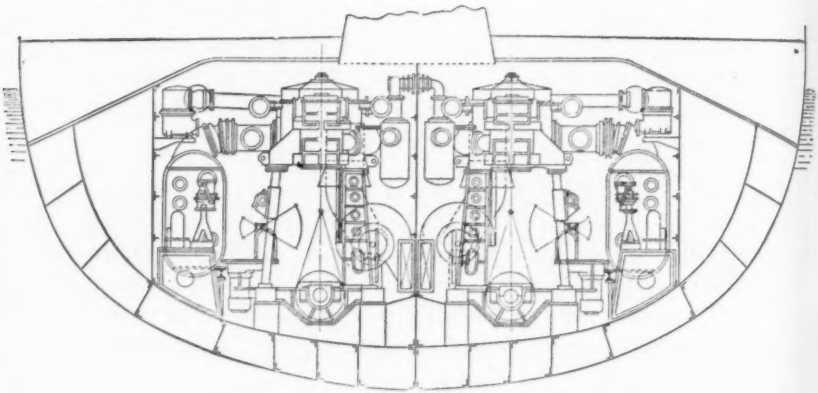
Length between perpendiculars, 435 feet; length over all, 474½ feet; breadth, 73½ feet; draught, 25 feet 9 inches; displacement, 12,625 tons.

The propelling machinery of the *Roma* was built by Messrs. Ansaldo, Armstrong & Co., at their Sampierdareno works.

The two main engines are four-cylinder, triple-expansion engines of the vertical inverted type, balanced according to the Yarrow-Schlick-Tweedy system. They are placed in two separate compartments, divided from each other by a central bulkhead, and are designed to develop 20,000 indicated horsepower at 125 revolutions per minute, with a boiler pressure of 210 pounds per square inch.

The diameters of the high-pressure, intermediate, and of each of the two low-pressure cylinders are, respectively, 39⅜ inches, 64½ inches and 74¾ inches. The stroke is 46 inches. All

the cylinders have steam jackets. The high-pressure and intermediate-pressure cylinders have piston valves, while the low-pressure cylinders have double-ported slide valves, with compensators at the back. Simple balance cylinders are fitted, both to the piston and to the slide valves. The valve gear is of Stephenson link-motion type. The reversing engine is of the all-round type. The pistons, cylinder covers, steam-chest covers, bed plates and columns are of cast steel, the pillars being of wrought steel.



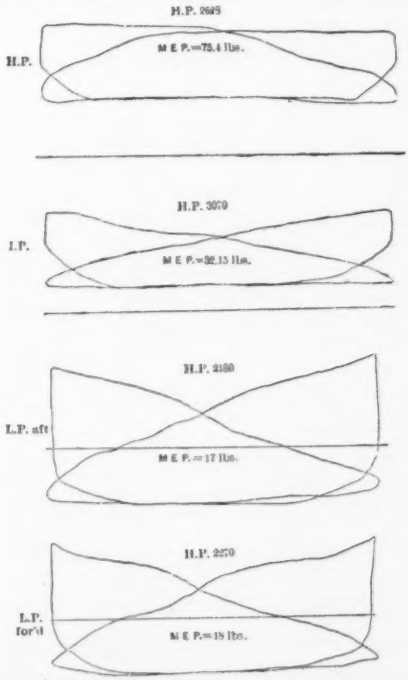
SECTION IN ENGINE ROOM, LOOKING FORWARD.

The arrangement of cylinders from forward aft is: Low-pressure, high-pressure, intermediate-pressure, low-pressure. The distance between the forward low-pressure cylinder and the high-pressure cylinder is $84\frac{1}{4}$ inches; the same distance is between the intermediate and the last low-pressure cylinder, while the distance between the high and the intermediate is 163 inches.

The crank shafts are of steel and hollow. Each is composed of two parts, and there are two cranks on each part. Each crank shaft has six main bearings, of 14 feet $2\frac{1}{2}$ inches total length. The outside and inside diameters of the crankshafts in the bearings are, respectively, $18\frac{3}{8}$ inches and $9\frac{5}{8}$ inches. The diameter of the crank pins is $19\frac{5}{16}$ inches, and their length

$22\frac{7}{16}$ inches. The turning wheel is fixed on the coupling at the after end of the engine.

The length of the connecting rods is 92 inches, and the minimum diameter of the solid connecting rods of the high and intermediate-pressure cylinders is 9 inches; the connecting rods of the low-pressure cylinders have the same diameter, but are hollow, the inside diameter being $5\frac{5}{16}$ inches. The out-

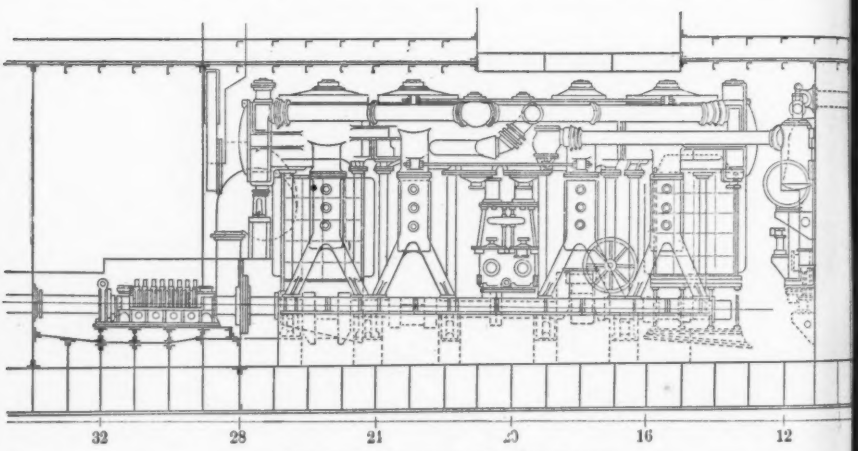


INDICATOR CARDS FROM THE ENGINES OF THE "ROMA." TOTAL HORSEPOWER, 10,148; REVOLUTIONS PER MINUTE, 124.8.

side diameter of the piston rods is 9 inches; the low-pressure rods are hollow, the inside diameter being 4 inches. The outside diameter of the pillars is 8 inches, and the inside diameter 4 inches. The main steam pipe is $13\frac{3}{4}$ inches in diameter, and the diameter of the pipe between the high and intermediate-

pressure cylinders is 20⅞ inches. Four 15¾-inch pipes bring steam from the intermediate-pressure cylinder to the two low-pressure cylinders. The diameter of thrust shaft is 18⅜ inches outside and 9⅝ inches inside. Each thrust shaft has eight collars, with a thrust surface of 17 square feet. The screw-shafts have a diameter of 18¾ inches. The propellers are of manganese-bronze with the three blades each. Their diameter is 17 feet 8½ inches; the mean pitch 20 feet 4 inches, and the developed surface 79½ square feet.

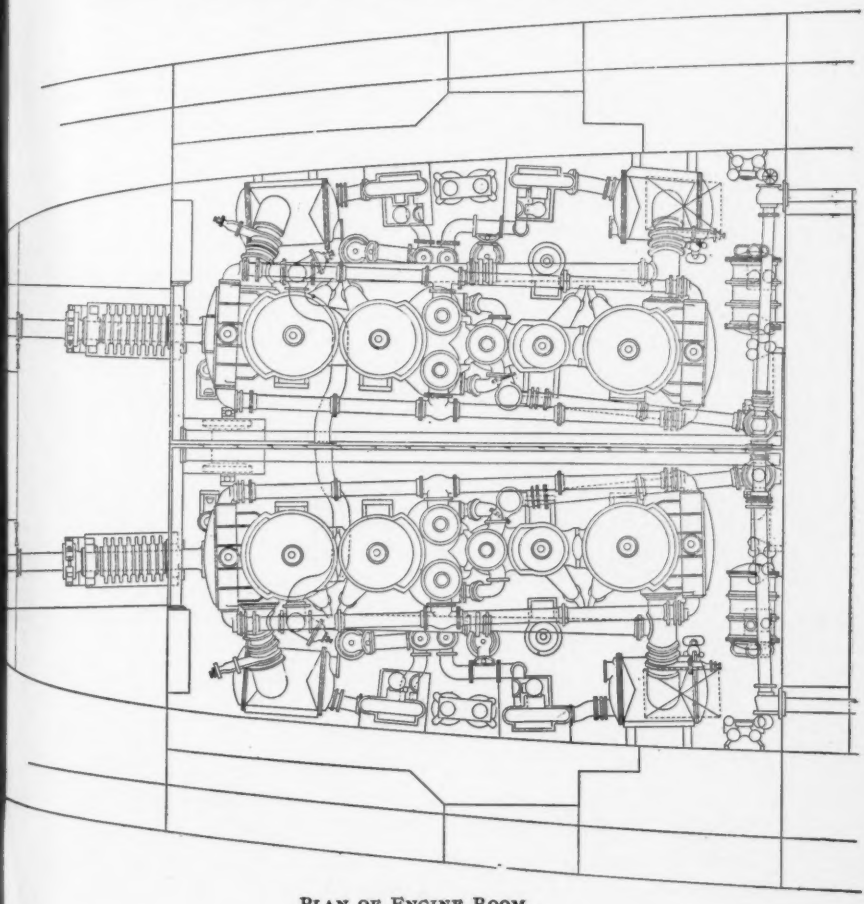
There are four main condensers, two for each engine room,



SECTION THROUGH ENGINE ROOM.

with a total cooling surface of 22,066 square feet. In each engine room the circulating water is supplied by two centrifugal pumps, driven by compound engines. Each pump is capable of delivering the circulating water to each of the two condensers independently or to both of them, and to draw from the bilges 1,000 tons of water per hour. There are in each engine room, also, two air pumps of the Weir type. One single air pump is also driven directly by each propelling engine from the high-pressure crosshead. There is also in each engine room an auxiliary condenser of 1,184 square feet.

The revolution counters are of the Molinari type. The other auxiliaries placed in the engine room are as follows: Two Ansaldo's evaporators complete with pumps, each capable of



PLAN OF ENGINE ROOM.

supplying 40 tons of fresh water during twenty-four hours; two independent bilge and fire pumps; two main, two auxiliary hotwell pumps and four fans.

There are eighteen Babcock & Wilcox water-tube boilers, arranged in three compartments, situated forward of the engine room. The after and central compartments each contain six boilers of twenty-two elements, and the forward compartment six boilers of nineteen elements. The total heating surface is 56,467 square feet, and the total grate surface 1,636 square feet. There are three funnels, having a total area of 252½ square feet. The height of the funnels above the grate-bars is 95 feet.

Two Weir feed pumps are placed in each boiler compartment, each being capable of supplying the water necessary for all the boilers in its compartment. Forced draft of the closed-stokehole system is obtained by means of twelve fans. There are, besides, six See ejectors with three duplex pumps for expelling the ashes, and six electrical ash hoists. The main and auxiliary steam pipes are of solid-drawn steel.

The first full-power sea trial took place between Spezia and Genoa, August 8. The speed measured while the ship was developing her maximum power (about 21,000 horsepower) was about 22½ knots, and the coal consumed 1½ pounds per horsepower.

We may add that the *Roma's* hull is of steel throughout. The armor, which is face hardened, is distributed as follows: A 10-inch waterline belt, extending the length of the machinery space, tapering to 4 inches at the ends; a central redoubt of 8-inch armor, and a 3⅛-inch protective deck, worked in forward. The large turrets have 10-inch armor, and the smaller turrets 6-inch armor.

She carries two 12-inch, twelve 8-inch, eight 3⅛-inch and twelve 1⅞-inch guns, and is provided with two submerged torpedo tubes. Her ordinary bunker capacity is 1,000 tons, and the maximum capacity 2,000 tons. The 12-inch guns are mounted in turrets, one forward and one aft. The 8-inch guns are mounted in pairs in turrets amidships. Four of the 3⅛-inch guns are mounted in barbettes, two at the bow and two at the stern, while the rest are located in various commanding positions, distributed over the whole vessel.—DAGINO ATTILIO, in "International Marine Engineering."

MANUFACTURER'S NOTES.

OXHYDRIC PROCESS.

The American Oxhydric Co., of Milwaukee, Wis., have prepared a very interesting illustrative circular describing the oxhydric process for the cutting and welding of metals.

The oxhydric torch is used for cutting the metal. There are two tubes in the torch, one using a mixture of hydrogen and oxygen which heats the metal, and the other using oxygen to cut the metal. By means of the torch the cut can be made to follow any desired line.

The oxhydric process uses the autogenous system of welding and the use of the oxhydric flame.

The applications of the process, as given in the circular, are as follows:

To cut steel and wrought iron to the thickness of from twelve to twenty inches.

To cut off rivet heads, quickly and easily.

To cut dies—also repair them.

To make all kinds of metal fluid and liquid containers without joints and less liable to leak when bruised or dented. Also to cut out defective parts and repair with a perfect sheet, the same shape, and weld same into position edge to edge.

To fill in holes drilled by mistake—dress them down and render them undiscernible.

To reclaim light and heavy castings coming from the sand with blow holes, sand holes, lugs off, and so on—also cracked or broken castings, whether of aluminum, brass, bronze, cast iron or cast steel.

To weld flanges on pipes.

To extend shafting, piping, rails, and so on, that have been cut too short.

To add metal to parts that have been subjected to friction, making those parts as serviceable as when first used.

To unite cast-iron heads to valve stems.

To renew teeth that have been broken from gear wheels.

It would appear that the very extended use of a process of this kind in ship building and ship and machinery repairs is a matter of very short time. In fact, processes of a similar kind are already being extensively used, and it appears that the oxhydric process enables the process of cutting and welding to be done with increased ease and rapidity.

STEAM TRAPS.

Two types of steam traps which appear to be well adapted to naval use are illustrated together with description and points of advantage claimed by their manufacturers.

*Lytton Steam Trap.**

Some Special Features of the Lytton Steam Trap.—(a) All of the working parts, with the exception of the bucket float alone, are on the outside of the receptacle, and therefore in sight and readily accessible. It is recognized that no trap can be absolutely infallible, but with this design any trouble can be immediately located and corrected with a minimum of time and cost without the necessity of disconnecting any piping or taking a bonnet off of the trap.

(b) Any of the automatic functions of the trap can be performed by hand from the outside; hence in the event of any solid matter lodging under the valve during discharge, it can usually be dislodged by simply submerging the float and allowing the trap to blow through for a moment.

(c) The discharge valve is practically balanced, and its operation is therefore independent of the pressure.

(d) The trap will discharge against any pressure less than that within the trap.

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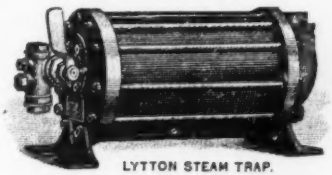
*Lytton Manufacturing Corporation, Franklin, Virginia.

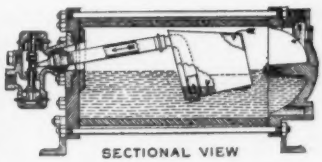
(e) The discharge valve being balanced, its size is not limited by the pressure, as in most traps, and a large valve opening is therefore obtained with a consequently increased discharging capacity.

(f) The float is of the open-bucket type, made of copper and brazed, and will last indefinitely.

(g) The copper gasket on the inlet head also forms a vertical strainer. A plug is provided on the under side of the inlet head for blowing off or cleaning out any sediment or other matter stopped by this strainer. The strainer being vertical and practically equal in diameter to that of the trap receptacle, there is little probability of its becoming stopped up to a sufficient extent to interfere with the inflowing water.



LYTTON STEAM TRAP.



SECTIONAL VIEW

(h) The trap being provided with detachable feet and adjustable heads drilled to template, it can be installed to sit on a deck, bolted directly to a bulkhead or partition or bolted directly to a deck overhead, thereby obviating all necessity for brackets, hangers, or any other support of such nature.

(i) The weight of the trap, for a given capacity and for a given strength, is considerably less, as a result of the patented construction used, than most traps on the market. The construction used also gives an apparatus of much neater appearance than any other trap on the market.

(j) All of the working parts are extremely simple in design and operation, and it is believed they are as nearly absolutely reliable as it is possible to obtain.

Operation.—When the trap is first installed it will be necessary to either fill the receptacle approximately one-half full of water, or to hold the hand lever (P) forward, which will raise the float (D), and allow the discharge valve (L) to re-

main closed, until sufficient condensation has flown from the system to the trap to support the float in position shown in cut.

When the above condition has been established the water flowing in the trap, through inlet opening (A) will raise the float (D) until it strikes the outer casing; when the trap has filled, the water will flow into float (D), causing it to drop to the bottom; this will cause an upward movement of the lever (K), which raises the auxiliary valve (M) and admits the pressure in the trap to the under side of the piston attached to main valve (L), which opens the main valve and the water flows out of trap through float (D), connection (Q), pipe (C) to outlet (B). The water will flow from the trap until it is level with the top of float, with float at lowest position, when almost instantly sufficient water has been blown from float, so that it will raise and float in the position shown in sectional view. This closes the discharge valve, which will remain closed until bucket float again fills and drops to the bottom.

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*The Squires Improved Steam Trap.**

The Squires trap gives an intermittent discharge and does so through a wide open valve, removing the water of condensation from steam pipes, steam separators, heaters, coils, stills, engine cylinders, receivers, etc. The valve is either closed tight or wide open, thereby eliminating the wire drawing on the valve and seat.

Note the convenience in gaining access to the valve and seat, which can be inspected or changed in a few minutes by unbolting the cap for any pressure from o upwards, including our unlimited valve mechanism, which is of great capacity.

The valves and seats in all our traps are made from the best bronze metal and have the figures cast on them, signifying the highest pressure under which the trap will operate. If at any time a trap should fail to discharge, this would indicate that the trap did not have the proper valve and seat for the pres-

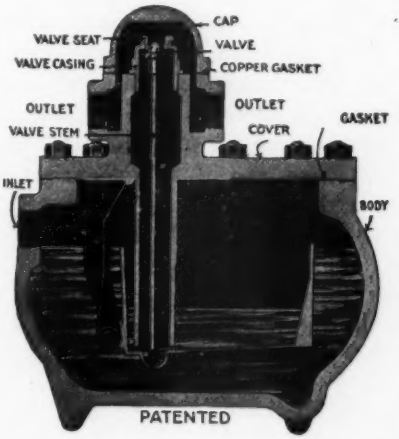
*C. E. Squires & Co., Cleveland, Ohio.

sure in the trap, and can easily be demonstrated by removing the cap and noting the figures on the valve and seat.

The Squires traps will discharge the condensation to any height corresponding with the steam pressure; one pound being necessary for each twenty-seven inches to lift desired.

The cast-iron bucket is hinged on bronze coned bearings, which cannot stick.

Operation of the Squires Improved Regular Trap.—When the condensation in the trap body has reached the high-water line, the bucket fills and sinks. The pressure at the trap then discharges the water through the valve port to the outlet, that is in the bucket together with the condensation in the trap body until it has reached the low-water line. The bucket then rises and closes the valve until the operation is repeated.



The Squires Improved Unlimited Pressure Steam Trap is a big capacity trap for high pressures; has all the merits of the Squires Improved Regular Trap, and in addition a valve mechanism which opens at lightning speed, enabling it to handle large amounts of water and work equally well at all pressures up to 300 pounds.

The big discharge valve is especially adapted for difficult

condensation problems. In large steam plants that have heavy condensation in the headers, or "slugs" of water from foaming boilers, this trap responds quickly to such floods and is always ready for a sudden repetition.

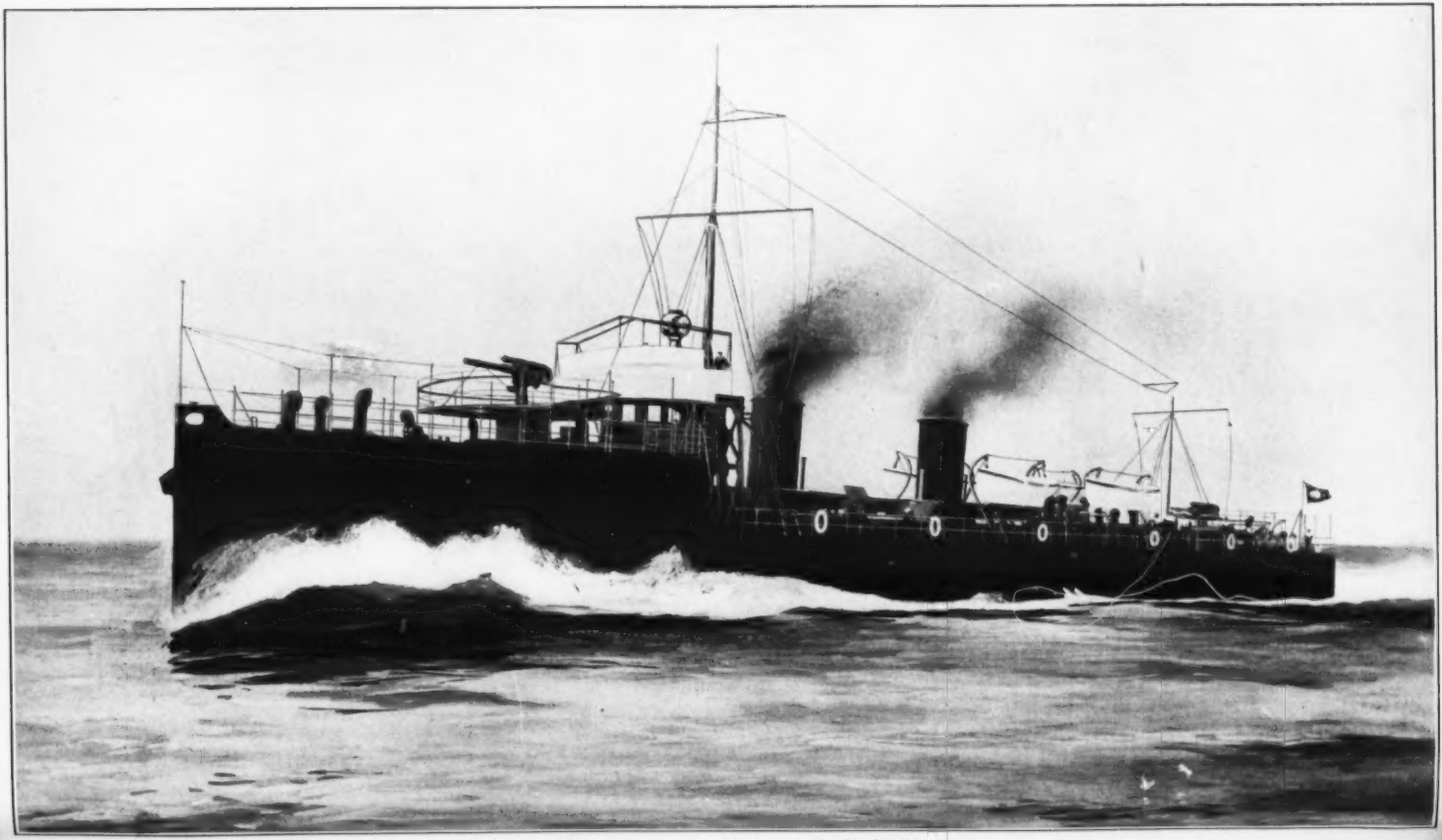
Operation of the Unlimited Trap.—When the condensation has reached the high-water line the bucket fills, sinks and opens the auxiliary valve. The pressure under the main valve is thus relieved, and the main valve is forced down and opens instantly by the internal trap pressure upon its upper surface. Nearly all the water in the bucket is then discharged together with the condensation in the trap body until it has reached the low-water line. The bucket then rises and closes both valves until the operation is repeated.

Directions for Connecting the Squires Improved Steam Trap.—This trap has two outlets and two blow-off connections; the pipe connections can be made on either side, whichever may be more convenient, leaving the other side plugged.

Blow out the trap as often as the condition of the water makes necessary. If the trap becomes clogged with mud, caused by boilers priming, the plug on the blow-off may be removed and the settling chamber can be cleaned out with a rod without disconnecting any piping.

The valve is attached to the bucket by a brass rod and is opened when the bucket tilts down.

Be sure after installing this trap that there is enough water in it to raise the bucket, as the valve does not seat until the bucket is up.



BRAZILIAN DESTROYER, BUILT BY A. F. YARROW.

SHIPS.

BRAZIL.

The trials of the torpedo-boat destroyer *Alagoas* have now been completed, this vessel being the seventh of the ten destroyers ordered by the Brazilian Government from Messrs. Yarrow & Company, of Glasgow.

The official full-speed trial took place on 26th November, on the Skelmorlie measured mile at the mouth of the Clyde, during boisterous weather, and the contract speed of 27 knots for three hours was easily obtained, carrying a load of 100 tons. The trial was under the supervision of Captain A. Rosauro de Almeida and Captain Bartholomeo Francisco de Souza e Silva, these gentlemen representing the Naval and Engineering Departments of the Brazilian Naval Commission in Europe.

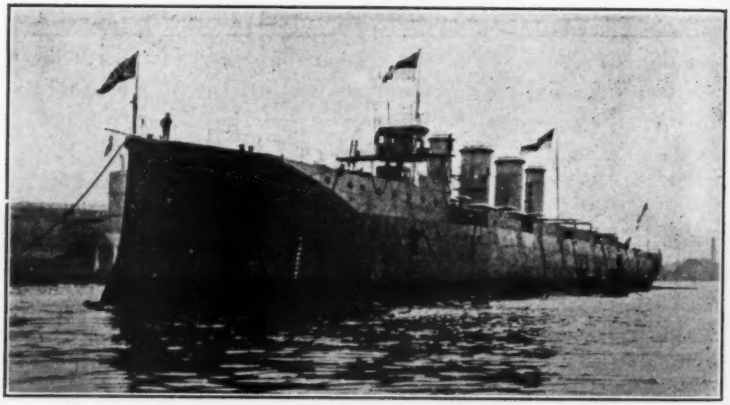
These vessels are 240 feet long by 23 feet 6 inches beam, propelled by two sets of four-cylinder reciprocating engines of 8,000 horsepower collectively, balanced on the Yarrow, Schlick and Tweedy system. Steam is supplied by two double-ended Yarrow boilers of the latest type.

Four of the ten destroyers, viz: the *Pará, Piauhy, Amazonas* and *Matto-Grosso,* have already arrived at Rio, and the fifth and sixth boats, the *Rio Grande de Norte* and the *Parahyba,* are on their way out to the Brazils. There are still three vessels under construction, viz: the *Santa Catharina, Paraná* and *Sergipe,* and these are nearing completion.

ENGLAND.

The New Scout Cruisers.—The launch, on the 25th of November, 1909, of H.M.S. *Newcastle,* brings the number of the new scout cruisers of the City class now afloat to four. The

first launched was the *Glasgow,* by the Fairfield Shipbuilding and Engineering Company, Limited, Glasgow; the second, the *Gloucester,* by Messrs. William Beardmore & Co., Limited, Dalmuir; the third, the *Liverpool,* by Messrs. Vickers Sons & Maxim, Limited, Barrow-in-Furness; and the fourth, the *Newcastle,* by Sir W. G. Armstrong, Whitworth & Co., Limited, Elswick. The fifth vessel ordered is the *Bristol,* which differs from the others in having Curtis, instead of Parsons, turbines; and, with characteristic caution, Messrs. John Brown & Co., Limited, Clydebank, the contractors for this ship, undertook an



H. M. Scout Cruiser "New Castle."

exhaustive series of tests with a new Curtis turbine installation before settling upon the details of design, so that if delay in delivery is involved, the resultant higher efficiency of the machinery will compensate for the postponement of the completion of the ship. This explanation is, in a measure, justified, as there is such a keen rivalry among the firms. The order of launching does not necessarily indicate the relative state of progress in construction.

Thus the *Newcastle* was unquestionably in a more advanced stage than any of the other ships at the date of launch. Indeed, no war vessel of this size has been launched with so much of

her machinery on board. As Sir Andrew Noble said at the lunch which followed the launch, there was placed on board before the launch the whole of the boilers, funnels, condensers, wing turbines and other auxiliary machinery, while the whole of the machinery and boiler-room bulkheads and the protective deck above the machinery and boiler compartments have been finally riveted, completed and water-tested. Only two heavy weights—the low-pressure turbines—remained to be lifted on board, and this operation was done on the next day.

All five ships are alike, excepting for the differences due to the machinery adopted, and have been built to the designs of Sir Philip Watts, K.C.B., the Director of Naval Construction, and to Admiralty specification. The principal dimensions are: Length between perpendiculars, 430 feet; breadth, extreme, 47 feet; mean draught, 15 feet 3 inches, with a corresponding displacement of about 4,800 tons. It is anticipated that a speed of about 25 knots will be obtained on trial with a shaft horsepower of about 22,000. The armament consists of two 6-inch breech-loading guns, one of which is placed on the forecastle deck forward, and the other on the upper deck aft; ten 4-inch breech-loading guns, five on each broadside, on the upper deck amidships; two Maxim guns, which are placed on the navigating bridge, and two 18-inch submerged torpedo tubes.

The vital parts of the vessel are protected by a nickel-steel armored deck, running all fore-and-aft, the sloping sides of which extend well below the water line. Additional protection is provided by the coal in the bunkers, which extend along each side of the vessel for the entire length of the machinery spaces above the protective deck, and for the length of the boiler rooms under the protective deck. The magazines and shell rooms, from which the guns are served, are ventilated and lighted on the latest system, and kept at a low, even temperature by special refrigerating plant installed for the purpose. "Ready-use magazines" with armored walls are introduced on the protective deck, having hand-up scuttles on the upper deck overhead, ad-

jacent to, and for the rapid supply of, the 4-inch broadside guns.

The vessel is subdivided into numerous watertight compartments; the turbine machinery which drives the four propeller shafts is contained in three separate engine rooms, with two additional compartments for the main condensers, while the twelve water-tube boilers are placed in three separate boiler rooms. The double bottom which extends under the machinery spaces and the forward magazines is also subdivided into numerous compartments which are used for carrying oil fuel and reserve-feed water for the boilers.

The vessel has four funnels and two masts, as shown in the view of the *Newcastle*. The masts are square rigged, suitably stayed, and arranged to take the gear in connection with the wireless-telegraphy installation and signaling apparatus, etc. A platform will be fitted on the fore lower mast, from which the firing of the several guns will be electrically directed. A long, raised forecastle is built on the upper deck forward, and the navigating bridge and compass platforms are fitted over the after end of the forecastle deck. Two powerful searchlights will be placed on the navigating bridge and the lower conning tower on the platform deck, an armored tube placing them in communication with one another.

A steam-driven capstan gear for dealing with the anchors is placed on the fore end of the forecastle deck, and an electrically-driven warping capstan on the upper deck aft. Two electric coal hoists will be fitted on the upper deck amidships, arranged for rapid coaling. The heavier ship's boats will be placed under davits and supported on skid beams, and the lighter boats will be hung under davits over the upper deck amidships. The ship's galleys, bakery and the refrigerating-machinery room, etc., are to be placed in deck houses in the casings on the upper deck amidships.

Accommodation for the captain and chief officers of the ship will be provided on the upper deck forward, inside the forecastle, and for the warrant officers, etc., on the protective deck

forward, while the crew's quarters, with their wash places, etc., will be placed on the protective deck aft.

Complete systems of electric lighting, ventilation and steam heating are arranged throughout the ship, and provision is made whereby the fresh air delivered to the living spaces can be heated if required.

As we have said, the propelling machinery of four of the vessels is of the Parsons turbine type, of 22,000 shaft horsepower, having four lines of shafting arranged in three separate engine rooms. The high-pressure ahead and separate high-pressure astern turbines are arranged on one shaft in each wing compartment. One low-pressure ahead turbine (in which is embodied the low-pressure astern turbine) is mounted on each of the inside shafts, being placed in a center compartment, which also contains the maneuvering gear for all the turbines. There are no separate cruising turbines, a special series of blades for cruising purposes being fitted in each high-pressure turbine. The propellers, four in number, of bronze, are arranged to run outward. At the aft end of the engine rooms, and separated from them by a watertight bulkhead, there are two auxiliary-machinery rooms, a center-line bulkhead dividing the one from the other. The main condensers, which are two in number, are arranged in these compartments, which contain also air pumps, circulating pumps, feed-water filters, evaporating and distilling plant, etc.

The boilers, twelve in number, of the small-tube type, are arranged in three separate boiler rooms, and in each boiler room there will be one main and one auxiliary-feed pump. The boiler rooms are adapted for use with the closed-stokehold system of forced draft, the supply of air being provided by steam-driven fans. Oil pumps are provided for both oil fuel and forced-lubrication purposes.

Under this year's program four other vessels of the same class have been ordered: one each from Sir W. G. Armstrong, Whitworth & Co., Limited, Messrs. Vickers Sons & Maxim, Limited, Messrs. William Beardmore & Co., Limited, and the London and Glasgow Shipbuilding Company, the last to be

fitted with Curtis turbines by Messrs. John Brown & Co., Limited.

Of somewhat similar type are the four cruisers of the *Boadicea* class, but smaller, being of 3,380 tons displacement and of 25 knots speed. The prototype, laid down in 1907, and the *Bellona,* begun a year later, have been completed; the *Blanche* was launched on the 25th November. All three were built at the Pembroke Dockyard, where a fourth is now in course of construction. These thirteen vessels—nine of the City class and four of the *Boadicea* type—go some way to strengthen the battle fleet with "eyes" and "ears;" but we can afford to have still more of these fast scouts.—"Engineering."

The Steam and Gun Trials of H. M. S. Vanguard.—The latest of our *Dreadnought* battleships, H. M. S. *Vanguard,* has completed her steam and gun trials, with most satisfactory results, and returned on the 26th November to the works of her builders, Messrs. Vickers Sons & Maxim, Limited, Barrow-in-Furness, there to be completed for commission. As she was only ordered in March, 1908, and her first keel plate laid on April 2, she has completed her trials within twenty months of construction being commenced, and promises to create a record for rapid battleship construction in private works, and in all establishments if the *Dreadnought* be excluded. It is notable, too, that the *Vanguard* has been constructed at a time when the Vickers Works had on hand another *Dreadnought* battleship for Brazil, a cruiser, submarine boats, and extensive merchant work; and in the engineering department machinery for five ships, besides gun mountings for several battleships for British and foreign Powers. It is the more important to bring out these facts at a time when so much is said in depreciation of our warship-building resources, since they prove to possible foreign clients that their orders can be entrusted to British firms with surety of delivery in the minimum time, as well as with certainty of the highest efficiency being achieved in all fighting elements.

The *Vanguard* is one of the three battleships of the 1907-8

program, but the Admiralty delayed the placing of the order in view of a probable attempt being made at the Hague Conference towards the minimizing of armaments; but this aim, as usual, was not realized. It is satisfactory to find now that although the *Vanguard* was not commenced until three months later than the two other ships, she will be completed earlier. The *St. Vincent*, which was laid down on December 30, 1907, will begin her trials on Monday, and the *Collingwood*, laid down on February 3, 1908, will proceed on her steam tests about a month hence. All three ships are of the same design, for which Sir Philip Watts, K.C.B., Director of Naval Construction, is responsible, and proof of high efficiency is found in the fact that the *Vanguard* exceeded the designed speed by 1 mile per hour. Each of the three ships has a length of 500 feet, a beam of 84 feet, and at 27 feet draught displaces 19,250 tons. The *Vanguard* is fitted with Babcock & Wilcox boilers, constructed at Barrow-in-Furness, supplying steam to four ahead or four astern turbines of the Parsons type, also made at the Vickers Works, to develop a total of 24,500 shaft horsepower through four shafts.

The program of trials included thirty-hour runs at 20 and 70 per cent. of the full power, eight hours at full power, and a series of tests, each of four hours' duration, at various powers, some with and others without the cruising turbines in use, and some with the closed exhaust and others with open exhaust. These trials, it is scarcely necessary to say, yielded most suggestive data, confirming results got by improvements incorporated in recent Admiralty practice and pointing the way to further developments. The closed exhaust on trials at less than half-power reduced the total steam consumption per shaft horsepower per hour by quite 20 per cent., which of itself is suggestive of the sound engineering policy being pursued at the Admiralty.

On the full-power trial, which was of eight hours' duration, the designed power was easily realized with the machinery running at about 330 revolutions per minute. Several runs were made over the measured course at Polperro, in order to

determine the speed of the ship, which was then loaded with water and other ballast to a draught slightly in excess of the service condition. Notwithstanding this, the mean speed realized was within a small fraction of 22 knots, or 1 mile per hour more than the designed speed. The economy realized by the machinery was high. The coal consumption per shaft horsepower was 1.63 pounds per hour, and the main steam consumption was under 14 pounds per shaft horsepower per hour. On the 30 hours' trial, at what is regarded as the continuous cruising speed, at about 70 per cent. of the total power, the results were equally satisfactory. Again, the vessel was loaded with ballast to exceed the designed displacement, but on the measured course she steamed at 19½ knots, while the machinery by torsionmeter showed that 17,600 horsepower was being transmitted by the shafts. On this trial the coal consumption was 1.7 pounds, and the steam consumption for all purposes 15.7 pounds per shaft horsepower per hour. On this trial all the auxiliary machines were at work, and the evaporators were used to make up even more water than was unavoidably lost, so that the coal and steam consumption represented the maximum necessary on service conditions. The performance of the ship under sea-going conditions was found to be very favorable in every respect. A trial was also made at one-fifth the total power, when the speed realized, according to the performance on the Polperro course, was 13½ knots, which was regarded as thoroughly satisfactory.

Opportunity was also taken of testing the facility of the turbine machinery aboard for quick maneuvering of the ship, while many of the other auxiliary engines on board were subjected to the usual contract tests.

The *Vanguard,* while at sea on these trials, carried out her gun trials. Like all of our recent *Dreadnought* battleships, she is fitted with a main armament of ten guns of 12-inch caliber and twenty guns of 4-inch caliber, and the firing of these guns proved also the satisfactory character of the work done in connection with the manufacture of gun-mounting machinery at the Vickers Works.—"Engineering."

H.M. S. Lion.—"The Engineer" says: She will be about 700 feet long between perpendiculars and 86 feet 6 inches in beam. She will displace about 26,500 tons, and will be propelled by turbines of 70,000 shaft horsepower. Her speed will be probably not far short of 30 knots. Her armament will, following the all-big-gun principle, consists of eight 12-inch guns in four barbettes, and all will be placed on the center line. As her secondary armament she will probably carry sixteen 4-inch guns.

Unlike the *Orion*, the *Lion* will probably have two tripod masts, and, in view of the fact that she carries no less than forty-two boilers, which have to burn coal enough to develop 70,000 horsepower, we have shown her with four funnels. It may be recalled in support of this that the *Mauretania* and her sister of approximately the same power have each four funnels, and that the *Invincibles*, which develop only 40,000 horsepower have three. Hence, even with four funnels, if the same cross-sectional area per unit is allowed the *Lion* would be underfunneled. There is, moreover, the fact that the use of four uptakes permits a more convenient arrangement of the boiler rooms; and, finally, that there is room for the four. The only argument against them is increased target area; but four funnels have been frequently employed in cruisers, and even six.

OBITUARY.

HORACE SEE.

Horace See, the well-known engineer and Associate Member of this Society, died on Tuesday, December 14, 1909. The funeral services were held at St. Peter's Church, at Fourth and Pine Streets, Philadelphia. Mr. See died of heart disease in New York.

Mr. See had a wide acquaintance among naval and marine engineers and shipbuilders, and was associated with the Newport News Shipbuilding and Dry Dock Co. for several years. During his later years he was a consulting engineer, with offices in New York. Mr. See was the inventor of numerous machines and marine devices, and was recognized as being one to whom the development of marine engineering and shipbuilding in this country owes much.

MR. GEORGE THOMPSON.

The death of Mr. George Thompson, Superintendent Engineer of the Cunard Steamship Co., Ltd., occurred on November 30, 1909.

Though quite a young man, Mr. Thompson has had a strenuous and distinguished career. He commenced his business life at Belfast as a premiumed apprentice in the Engineering Works of Messrs. Harland & Wolff. His apprenticeship completed, he gained sea-going experience, serving, among others, on the steamers of the Lord Line. On gaining his extra Chief Engineer's certificate he returned to Messrs. Harland & Wolff's and entered their drawing office. Later he was entrusted with the important duty of supervising the erection of engines for

new vessels. Then he became Works Manager, and in June, 1903, he left Messrs. Harland & Wolff's, and passed into the service of the Cunard Steamship Co., Ltd., when Mr. Little held the position of Superintendent Engineer under Mr. James Bain, who recently retired from his position of General Superintendent to the Cunard Steamship Co. Mr. Thompson afterwards became Superintendent Engineer, on the retirement of Mr. Little, in September, 1904, and as such has achieved good work for the company. It will be remembered that the *Caronia* made her first sailing in February, 1905, that she was followed by the *Carmania* propelled by turbine engines, and until the advent of the *Lusitania* and *Mauretania* this 20,000 tons steamer was the largest turbine steamer in the world. With the engineering equipment of the *Carmania*, and later of the Cunard Express steamers, Mr. Thompson was closely associated, and it was always a source of gratification to him that the many problems connected with the building of 32,000-ton steamers and propelling them by turbine engines of close upon 70,000 horsepower were so successfully solved that the ships were enabled to make regular passages at an average speed per hour of over 25 nautical miles.

In engineering circles, both in Liverpool, Belfast and elsewhere, Mr. Thompson was justly regarded as right in the forefront of his profession on all matters pertaining to the propulsion of steamships, whether by turbine or reciprocating engines; refrigerating, electric lighting, ventilation and heating, and steam boilers, etc. But with his expert knowledge and experience he was always retiring, and hence did not loom so largely in the public eyes as he might have done.

ASSOCIATION NOTES.

At the annual meeting of the American Society of Naval Engineers, held at the Navy Department, Washington, D. C., December 31, 1909, the following were elected officers of the Society for the ensuing year:

President, Engineer-in-Chief H. I. Cone, U. S. Navy.

Secretary and Treasurer, Lieutenant Henry C. Dinger, U. S. Navy.

Council, Commander Wm. Strother Smith, U. S. Navy; Engineer-in-Chief Charles A. McAllister, U. S. R. C. S.; Lieutenant Commander J. K. Robison, U. S. Navy.

ANNUAL STATEMENT, JANUARY 1, 1910.

Received 1909.		
Dues and subscriptions	$4,780.03	
Advertisements	2,152.92	
Sales	739.24	
Interest	326.57	
Subscriptions for banquet	375.00	$8,373.76
Balance (cash) January 1, 1909		3,598.30
		$11,972.06
Expended.		
Printing and stationery, postage	*$5,379.66	
Engraving	628.07	
Drafting	42.00	
Salary, Secretary-Treasurer	900.00	
Employment of clerk	220.00	
Articles paid for	256.00	
Safe deposit box	5.00	
Refund on check	3.45	
Expenses of Society banquet	836.60	8,272.78
Cash balance		$3,699.28
Balance in bank, as per book	$3,609.28	
Cash on hand	90.00	
Cash balance		$3,699.28

* Of this amount, $861.08 was spent on the November, 1908, number, which was not issued till January, 1909. Allowing for this expenditure, which was made for the publication of the previous year, the net gain in cash for the year is $962.06.

Available assets:		
Cash balance, January 1, 1910	$3,699.28	
Six (6) $1,000.00 bonds, at $900.00	5,400.00	
Total		$9,099.28
Cash balance, January 1, 1909	$3,598.30	
Cash balance, January 1, 1910	3,699.28	
Net gain for year		$100.98

This statement was audited by a committee of the Council and found to be correct.

H. C. DINGER,
Lieutenant, U. S. Navy,
Secretary-Treasurer.

H. I. CONE,
Engineer-in-Chief, U. S. Navy,
President.

The membership is as follows:

Honorary members,	6.
Prize essayists,	8.
Members,	326.
Associates,	299.
Subscribers,	248.
Exchanges,	70.
New Members during year,	58.
Losses by death and resignation, Members,	15.
New Associates during year,	13.
Losses by death and resignation, Associates,	8.

All Members and Associates are specially encouraged to bring the desirability of membership in the Society to those who are eligible.

BY-LAWS OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.

1. The Association shall be known as the AMERICAN SOCIETY OF NAVAL ENGINEERS.

2. The object of the Society shall be to promote a knowledge of marine engineering and naval architecture by reading, discussing and publishing papers on professional subjects; by bringing together the results of experience acquired by engineers in all parts of the world, which, though valueless when unconnected, tend much to the advancement of engineering when published together in the JOURNAL of the Society; by publishing the results of such experimental and other inquiries as may be deemed essential to the advancement of the science; and historical events in the lives of engineers.

3. The officers of the Society shall be a President, a Secretary-Treasurer, and a Council, all of whom shall be elected annually.

4. The Society shall be composed of Members, Associates and Honorary Members.

5. Officers of the Line, Construction and Civil Engineer Corps, and ex-officers of the Line, Engineer, Construction and Civil Engineer Corps of the Navy, and officers and ex-officers of the Revenue Cutter Service shall be eligible as Members.

6. Persons in civil life whose knowledge of engineering is such that they can coöperate with naval engineers in the promotion of professional knowledge, or who are intimately connected with the engineering profession, shall be eligible as Associates. Commissioned Chief Warrant Officers of the Navy shall be eligible as Associates.

7. The Secretary of the Navy, the Assistant Secretary of the Navy, the Chief of the Bureau of Steam Engineering of the Navy Department, all ex-Chiefs of the Bureau of Steam

Engineering, the prize essayist of each year, and such other persons as the Society may elect, shall be Honorary Members.

8. Members shall be admitted upon application and payment of the annual subscription.

9. An Associate may be admitted upon application in writing and payment of the annual subscription, provided his application have the recommendation of a member, and receive the approval of a majority of the Council.

10. Associates shall be entitled to all the privileges of Members except voting and holding office.

11. To be eligible as Honorary Member the candidate must receive the unanimons vote of the Council before his name shall be presented to the Society; and the favorable vote of two-thirds of the Members voting shall be necessary for election.

12. The direction and management of the affairs of the Society, and the editing and publishing of the JOURNAL, shall be vested in a Council composed of five members, the President and the Secretary-Treasurer being members *ex officio*. The Council may appoint one of its number Librarian.

13. The President shall exercise the usual duties of that office.

14. The Secretary-Treasurer shall conduct the correspondence of the Society and its financial transactions. He shall submit an annual statement of the receipts and expenditures of the Society, which shall be audited by three other members of the Council.

15. The Council shall have authority to fill vacancies which may occur during the year.

16. Subscriptions and all matter intended for publication, shall be sent to the Secretary-Treasurer; but no paper shall be read before the Society at the annual meeting, nor anything published in the JOURNAL, without the approval of the Council.

17. In deciding matters pertaining to the Society none but members shall be entitled to vote; to constitute a quorum for business nine members must be present.

18. The Society shall meet annually in the City of Washington, and at such other times and places as meetings can be conveniently arranged.

19. The annual subscription shall be five dollars, payable in advance.

20. Copies of the JOURNAL may be sold by the Society at a price which shall be fixed by the Council.

21. Absent members voting on any proposition must state over their signatures whether they are for or against the proposition.

22. On the first Tuesday in October a meeting of the Society shall be held for the consideration of nominations to offices falling vacant under the rules. Nominations to office must be made in writing, and may be made by any member, whether present at the meeting or not. The consent of the nominee to accept office if elected must be assured. A necessary qualification of all nominees, besides full membership, is residence in or near Washington during the term of office.

23. After the nominations are all made, lists of the candidates for the different offices shall be prepared by the Secretary-Treasurer, and mailed to all members of the Society. Each voting list must be signed, and must show under the different offices the name of the person voted for. Each voter will enclose his ballot in a sealed envelope, which will be sent to the Secretary-Treasurer previous to the annual meeting. At the annual meeting the envelopes will be opened and the votes counted.

24. Nothing in the preceding section shall preclude the name of any candidate from being put in nomination for more than one office. In case of a tie the presiding officer shall decide. Voting by proxy is prohibited.

25. In the event of a candidate receiving the greatest number of votes for more than one office he shall decide which he will accept.

26. Any office falling vacant in the course of the year owing to the death, resignation or removal from Washington

or vicinity of the incumbent, shall be filled by vote of the members of the Council remaining.

27. Motions to amend these By-laws, or to make new By-laws, must be in writing, and must lie over at least three months, during which time notice thereof shall be sent to all members; and the assent of two-thirds of the members voting shall be necessary to amend or adopt.